# WOOD GASIFIER/GAS TURBINE DESIGN

# AND FEASIBILITY STUDY

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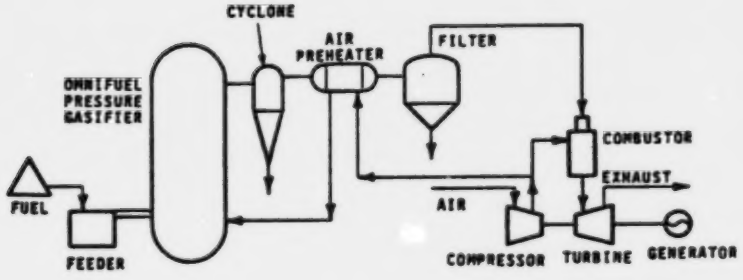
# E X E C U T I V E  S U M M A R Y

## EXECUTIVE SUMMARY

This study deals with technical and economic factors related to the production of a combustible gas from wood or peat and the utilization of that gas in a gas turbine for the production of electricity.

As can be seen in the accompanying flow schematic (Figure 1), the wood or peat fuel is fed into a pressurized Omnifuel fluidized bed gasifier where it is converted into a fuel gas. The pressurized gas is then cooled slightly and cleaned of ash contaminants before being combusted in a gas turbine generator set, in much the same way as oil or natural gas is used in gas turbines.

FIGURE I
GASIFIER/GAS TURBINE COMBINATION




Some air from the turbine air compressor can be utilized in the gasifier to complete a compact and efficient energy generation package. To this must be added a conventional wood or peat unloading, storage and drying system and a waste heat boiler system to recover energy from the turbine exhaust.

The major findings of this report are:

1. Gas turbines are available that would utilize the gas produced by an Omnifuel gasifier. Major turbine vendors have experience in burning similar fuels and are enthusiastic about this development. The Westinghouse 191 machine manufactured in Hamilton, Ontario, offers an ideal opportunity for a fully Canadian package.

2. Whilst there are many types of high temperature gas cleanup equipment under development and testing, none have a proven track record under the conditions required for this system. Many show promise, and a demonstration of the technology using one or more of the cleanup methods would be necessary.

3. The combined cycle systems (gas turbine plus steam turbine utilizing the waste heat) show overall electrical generation efficiencies between 25 to 30 percent are possible from wet wood in sizes from 20 to 50 MW electricity generation. This compares with 20% for a single steam cycle and 23% for a complex (multi-pressure) steam cycle. Complex cycles are not generally used for sizes less than 50 - 100 MW. If some of the more efficient gas turbines now operating on oil and natural gas are adapted for low BTU gas, even higher efficiencies are possible.

4. Capital costs would be between $800 and $1,000/kW installed capacity. This compares with $1,000 to $2,000 for conventional steam thermal power plants for sizes between 20 and 50 MW.

   Such a system would generate an internal rate of return of about 18% before tax in constant dollars under cost scenarios now applicable in many parts of the United States.

   In Canada the system is only economic if the steam produced from the waste heat is utilized for process heating to replace steam produced from natural gas or oil, as in a pulp and paper mill or if used on a smaller scale in a remote community to replace electricity generated from diesel fuel.

5. A demonstration at a small size (1.5 MW) would cost some $5 million, including two years operating and testing. Alternately, a larger system could be installed to operate on natural gas and provide a modest return on capital whilst a gasifier is demonstrated and phased into operation on the turbine thus minimizing the risk.

# INTRODUCTION

## INTRODUCTION

Omnifuel Gasification Systems is an Ontario-based company specializing in the development of large industrial fluidized-bed gasifiers of biomass materials. Based on prototype work carried out in the seventies, the company has built in Hearst, Ontario, what is believed to be the world's largest gasifier of wood waste in a commercial application. This installation has now been in operation for nearly two years.

Gasification has certain advantages over combustion devices when an inflammable gas is desired. For example, an existing steam boiler fired with oil or natural gas can be retrofitted for wood gas firing at much less cost than a complete rebuild of the boiler for wood firing. Most experts believe, however, that the true future of gasification lies in applications where combustion devices cannot compete at all. These include the direct firing of the gas in an internal combustion engine and the production of "syngas" which can be chemically transformed into other materials such as liquid fuel - e.g. methanol.

Early in 1982, Omnifuel completed a report for DSS/NRC entitled "A Design Study for a Wood Gasifier/Engine Test Project" which examined the coupling of a gasifier to various kinds of reciprocating engines. Such prime movers are generally used for electricity generation in non-interconnected systems, and when used with a gasifier offer a valuable energy alternative in locations where a biomass fuel is available.

The present report is a parallel study on the coupling of a gasifier with a gas turbine. Under certain economic conditions this combination may be the preferred route. The high exhaust temperature from a gas turbine permits a combined cycle to be designed which will have a much higher thermal efficiency overall than any electricity-only system.

It is the purpose of this study to identify existing equipment suitable to this task, to determine the technical feasibility

and efficiencies of such a package, and to establish the economic parameters in order to permit informed decisions on the potential for this technology.

Much of the equipment to be used in such a plant has been proven in similar duty. Gas turbines, for instance, have been in service since 1949 on blast furnace gas. Combustors to burn 3.5 MJ/Nm3 gas were under development as early as 1953 and to date almost 50 machines are operating on low heating value furnace or process exhaust gases.

Considerable research has been undertaken to accurately define upper limits of contaminant and particulate levels in the fuel gas. These areas are addressed in this study, based on typical gasifier feedstock, as well as the potential impact on the turbine components.

Control of these gaseous components is discussed under gas cleaning equipment. This section deals with what can be expected in the gas stream as well as a discussion of current and planned equipment available for gas clean-up.

The description and pricing of the gasification section incorporates the latest information available on pressurized fluid-bed gasifiers. Wherever possible, experience gained on existing projects is used as a basis for identifying any potential problems and forms the basis for practical solutions. This input is instrumental in determining a technically feasible gasification package.

Using budgetary prices for existing equipment and estimates for other items, composite price levels are established for various package sizes. These prices are then inserted into an economic model which demonstrates the financial merits of the proposed system.

# SYSTEM CONFIGURATION

## 2. SYSTEM CONFIGURATION

When utilizing conventional fuels, a gas turbine compresses air, injects and combusts a pressurized fuel into the air stream and then expands the heated air and combustion products in a turbo-expander.

Fuels that can be readily compressed, are relatively clean and have high heat content such as distillate oil or natural gas are conventionally and readily used in gas turbines. Relative to other forms of power generation equipment, the capital cost of gas turbines is low, but their requirement for high quality fuel and their lower efficiencies in the past have limited their use to peaking and standby power generation.

However, as turbine efficiencies have improved and the tolerance for dirty fuels increased, they have become more and more common in applications from pipeline gas compression to residual oil firing and of course as jet engines.

Solid fuels have to date, for the most part, eluded utilization in gas turbines. This is primarily because with the high ash content of the solid fuel, the products of combustion must be cleaned before introduction to the turbo expander. This is difficult at the high temperatures involved.

By first gasifying the solid fuel to produce a combustible gas, the relatively lower volumes of gas can be cleaned of ash at a lower temperature before the final combustion of the gas in the gas turbine. If the gasifier can be run under pressure thus elimating the need for a gas compressor, the optimum use of solid fuels in a gas turbine is achieved.

Figure 2 is a process schematic showing the combination of a pressurized Omnifuel Gasifier with a gas turbine. If necessary the fuel is first dried before being pressurised and fed to the gasifier. The product gas is then cooled slightly in a heat exchanger, filtered of particulate matter and fed to the combustion chamber(s) of the gas turbine.

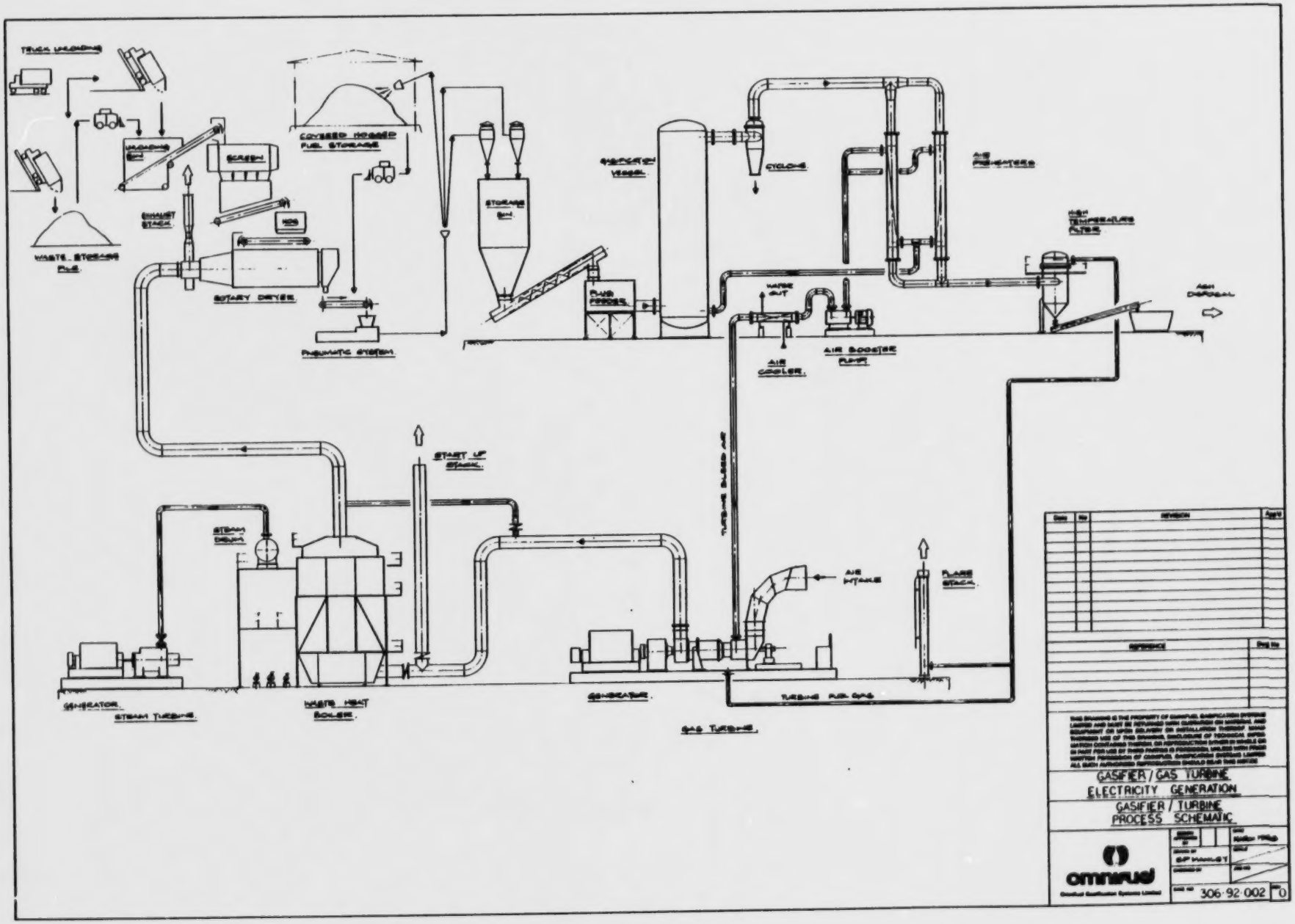

TRUCK UNLOADING
COVERED HOPPER FUEL STORAGE
SCREEN
EXHAUST STACK
WASTE STORAGE PILE
ROTARY DRYER
PNEUMATIC SYSTEM
STORAGE
FUEL FEEDER
GASIFIER VESSEL
CYCLONE
AIR HEATERS
HOT TEMPERATURE FILTER
ASH DISPOSAL
WATER OUT
AIR COOLER
AIR BOOSTER PUMP
START UP STACK
STEAM DRUM
TURBINE BLEED AIR
AIR INTAKE
FLARE STACK
GENERATOR
STEAM TURBINE
WASTE HEAT BOILER
GENERATOR
TURBINE FUEL GAS
GAS TURBINE
GASIFIER/GAS TURBINE ELECTRICITY GENERATION
GASIFIER / TURBINE PROCESS SCHEMATIC
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FIGURE 2

Because turbo compressor/expander combinations have been developed to utilize conventional high heating value fuels such as natural gas, with the higher mass flow of lower heating value fuels (such as the gas produced in a gasifier) it is usually necessary to bleed some of the air from the compressor stage to avoid overloading the expander. As pressurized air is required for the gasifier, this bleed air can thus be utilized as shown in the process schematic where it is first preheated before being introduced to the gasifier.

PROCESS DESIGN

Three different plant sizes have been chosen for more detailed evaluation. These are nominally in turbine generator output sizes of 20 MW, 30 MW, and 40 MW. It was originally intended to include a turbine in the 5-10 MW range but as no turbine was encountered at this size that was sufficiently developed for operation on low heating value gas fuels, this size was omitted and a larger 40 MW machine included. In addition, a small 1.5 MW system has been designed and priced as a possible demonstration project.

As each turbine has very specific operating parameters including operating pressure, turbine expander inlet temperature, compressor air flow, exhaust gas flow and exhaust gas temperature, actual turbines have been chosen at these sizes and the complete system designed around the requirements of the particular turbine.

The first step in the design is the characterization of the gas produced by the gasifier. Although several different feedstocks are contemplated for this type of system including wood, peat, biomass waste and even refuse derived fuel, wood has been chosen as a typical fuel. Other fuels do not vary significantly in terms of composition and the heating value of the product gas.

Green wood chips, wood waste or bark usually contain some 50% moisture (wet basis) and in order to obtain a gas that can be utilized in the turbine without too much difficulty, the wood must be dried before being fed to the gasifier.

The optimum wood moisture content after drying is approximately 30%. (See Section 7.1 - Cycle Efficiencies).

Omnifuel's computerized model was used to determine wood requirements, gasifier air requirement, and to characterise the product gas.

For wood of 30% moisture, the following gas is expected:

| | | |
|---|---|---|
| Analysis | CO | 12.3 vol % |
| | H2 | 8.4 |
| | CH4 | 4.5 |
| | CxHy | 2.0 |
| | N2, H2O, CO2 | Balance |
| Higher Heating Value | | 5.7 MJ/Nm3 |
| Lower Heating Value | | 5.2 MJ/Nm3 |
| Temperature (after cooling) | | 425C |
| Pressure | | as required |

Turbine manufacturers were consulted on the utilization of this gas in their machines, and on performance, etc.

It must be emphasized that it is desirable to use the gas hot at 425C or higher as below this temperature it is possible that some of the small quantities of condensible hydrocarbons may deposit in the filters, piping, valves, etc. At any lower temperature, the gas would have to be scrubbed with water to remove these organic components resulting in the loss of sensible heat in the gas and the addition of a liquid effluent stream for disposal.

The ability to run the gasifier under pressure eliminates the need for a gas compressor and therefore reduces the power requirements of the system and the need for a liquid scrubbing and gas cooling system.

The quantity of gas equivalent to the turbine manufacturers' specified heat rate is then combusted with turbo-compressor air such that the turbine's specified exhaust mass flow and turbine inlet temperature are matched. This would then determine the quantity of compressor air that must be bled from the system in

order to avoid overloading the turbine. This bleeding of air is necessary because all the commercial turbines available are matched on conventional fuels with higher energy density than the low heating value gas. Thus the extra mass of low heating value gas must be compensated for by bleeding air. Some of this bleed air is utilized, however, in the gasifier after intercooling and boosting to the pressure desired for the operation of the gasifier.

It is possible in certain machines to eliminate the need for bleed air if the wood is dried further to produce a higher gas heating value. The desirability of doing this would depend on site specific requirements and would need to be evaluated as detailed engineering for a specific project proceeds.

The turbine exhaust flow and temperature are then used to determine the size of the waste heat boiler system and the needs of the dryer.

# F U E L  G A S  P R O D U C T I O N

3.1 Wood Receiving and Preparation

3.2 Gasification System

## 3.1 Wood Receiving and Preparation

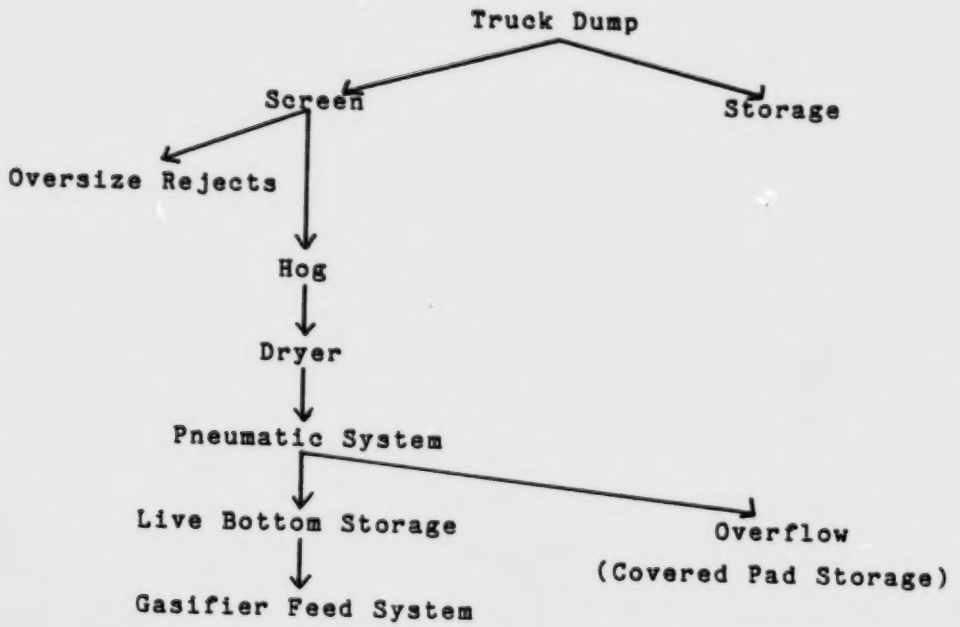
The gasifier must be fed minus 5cm fuel at a reasonably uniform moisture content on a continuous basis.

The fuel would be received on an 8 hour/day basis.

The equipment involved to meet these conditions includes: truck dump, pad storage, screen, hog, drier, live bottom storage, covered pad storage and front end loader. (See Figure 3)

FIGURE 3

WOOD PREPARATION FLOW




The volumes of wood involved are substantial in the cases considered.

There are large storage/reclaim systems available, but it is felt that the increased level of automation does not necessarily

increase the reliability of the feed and the cost is significantly higher.

| | m3/hr | m3/day |
|---|---|---|
| System 1 | 87 | 2,100 |
| System 2 | 136 | 3,260 |
| System 3 | 153 | 3,680 |
| Demonstration | 10 | 230 |

## 3.2 Gasification System

The major equipment items in the gasification System are:

Surge Bin
Weigh Feeder
Splitter
Plug Feeder
Startup System
Gasifier Vessel
Primary Cyclone
Gas Cooler
Letdown Valve
Flare Stack
Air Cooler
Booster
Instrumentation

A General Arrangement of the Gasifier is shown in Figure 4.

### SURGE BIN

This would be a live bottom storage device receiving material from the wood preparation. Its purpose would be to meter fuel to the gasification system. The type of live bottom would depend on the size of the installation, and the climate. It could be reciprocating flow, travelling screw, or a full bed of screws.

### WEIGH STATION

The weigh station would monitor the output of the surge bin and control its rate to match the demands of the gasifier controls.

### SPLITTER

The cases that were looked at were sufficiently large as to require multiple feed points into the gasifier. The splitter would divide the flow between the feed points. The split need not be absolute and it was determined that a volumetric device would be adequate. A small live bottom unit with levelling,

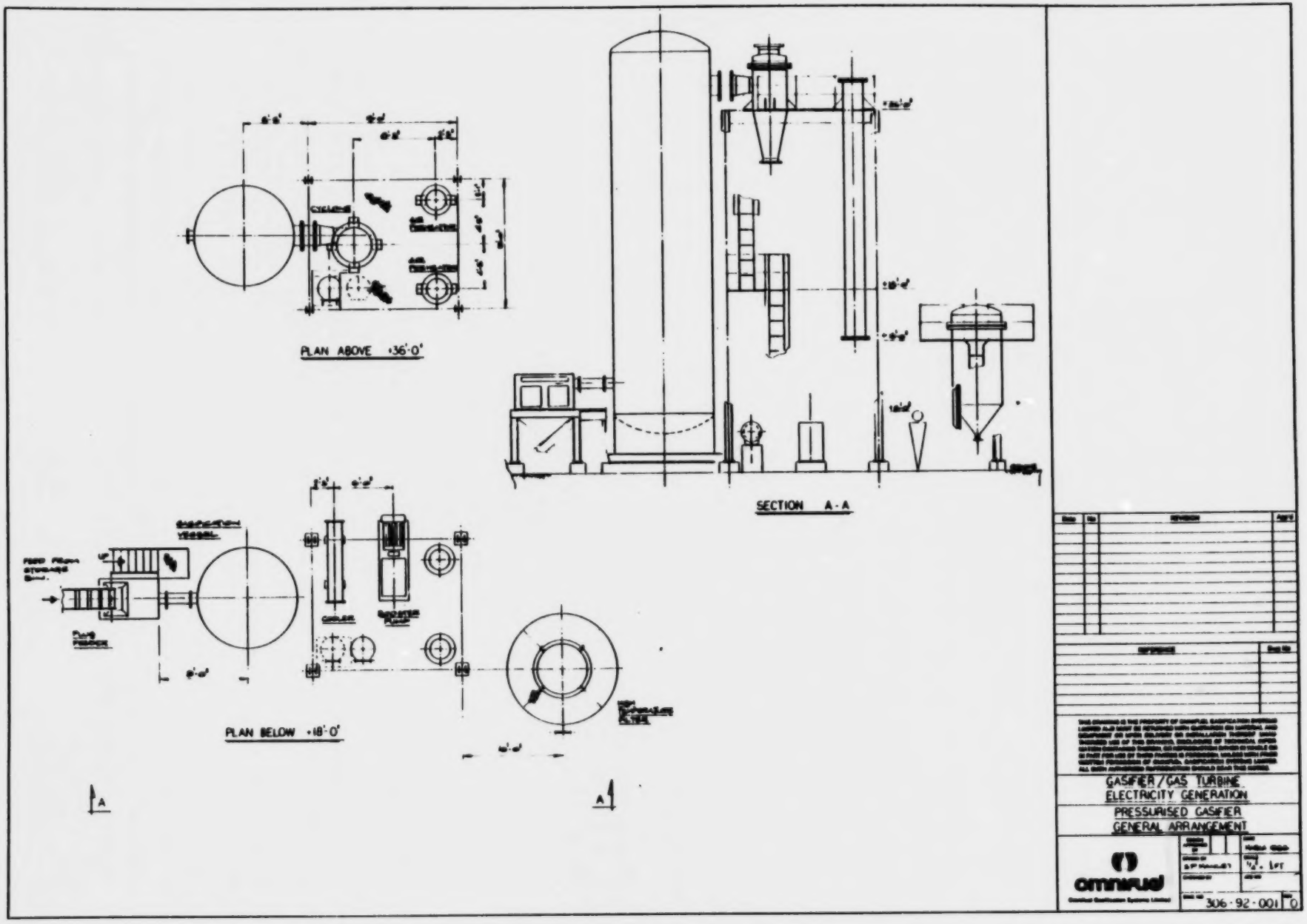

PLAN ABOVE +36'-0'
SECTION A-A
PLAN BELOW +18'-0'
GASIFIER/GAS TURBINE
ELECTRICITY GENERATION
PRESSURISED GASIFIER
GENERAL ARRANGEMENT
omnifuel
306-92-001

FIGURE 4

discharging into multiple feed conveyors will meet the requirements.

FEED CONVEYORS

The feed conveyors will take the material from the splitter to the plug feeders. These would be trough belts or screw conveyors depending on capacity and layouts.

PLUG FEEDER(S)

The plug feeder is the device which pumps the fuel into the pressurized fluid bed. The cases studied involve pressures as high as 2,000 kPa. The design selected for this service has also been operated commercially for several years at pressures up to 3,000 kPa. The feeder has also been operated commercially feeding a fluid bed gasifier at temperatures up to 840C. After an initial period of debugging, the feeder has provided two years of satisfactory performance.

Testing of the feeder under simulated conditions produced results which increase the level of confidence in this critical piece of equipment.

STARTUP SYSTEM

This is the system which would be used to bring the fluid bed gasifier up to operating temperature. The warm up need not be performed at elevated pressure and a low pressure blower and startup heater may be used. This will save considerable capital and operation cost. This equipment is then isolated from the gasifier to allow use of high pressures.

GASIFIER VESSEL

The gasifier vessel will be a mild steel pressure vessel internally lined with refractory and insulation. The reactor internals will be quite similar to a lower pressure fluidized bed with a few modifications to account for the increased gas density and reaction rates. As the temperature in the gasifier never exceeds 840C, long refractory life is realized without exotic materials.

## PRIMARY CYCLONE

A primary cyclone is included to improve system efficiency and keep large particles from damaging downstream equipment. This cyclone would be mild steel refractory lined and insulated. The wear areas of the cyclone will be lined with high grade refractory material.

## AIR PREHEATER

This unit cools the gas and preheats the fluidizing air. For simplicity of design, operation and ease of cleaning this will be a vertical shell and tube design of stainless steel construction.

## FLARE STACK

For startup and pressure trim a flare stack will be used. This flare stack will be of conventional design with a flare pilot and flame front ignition.

## BOOSTER

The bleed air from the turbine is a convenient pressurized air stream to use as fluidizing air. This stream must undergo further compression in a booster.

The characteristics of the booster are summarized below:

| System | Pressure (kPag) Inlet | Outlet | Ratio | Inlet (cu.m/hr) | kW |
|---|---|---|---|---|---|
| 3 | 1,310 | 2,033 | 1.52:1 | 4,050 | 930 |
| 2 | 723 | 1,412 | 1.8:1 | 6,100 | 1,230 |
| 1 | 620 | 1,137 | 1.7:1 | 4,350 | 670 |
| Demonstration | 276 | 551 | 1.7:1 | 2,930 | 270 |

Three types of compressors could be used in this service.

Multistage Centrifugal

- most expensive
- consistent with other types of machinery in plant
- compact
- approx. $500,000 for Westinghouse 191 service

Reciprocating

- large
- rugged
- simple
- approx. $425,000 for Westinghouse 191 service

High Speed Turbine

- compact
- requires multiple units (4)
- least expensive
- approx. $250,000 for Westinghouse 191 service

LETDOWN VALVE

The pressure of the gas needs to be reduced to near atmospheric to enter the flare stack system. The valve for this service will be selected carefully. A multiple orifice type valve of high alloy materials is likely to provide the best service.

AIR COOLER

The air cooler will be a shell and tube design with water flow on the shell side and the compressed air flowing through the tubes. Shell is contructed of mild steel and tubes of copper.

The air will be cooled to 80C using water entering at 20C and leaving at 75C.

CONTROL AND INSTRUMENTATION

General: The wood preparation, gasifier, gas cleanup, gas turbine and waste heat boiler all interact with each other and overall control is essential. A preliminary piping and instrument diagram for the gasifier and gas cleanup system is shown in Figure 5.

It is suggested that a distributed control system linked to a central control would be ideal. Each unit, especially the gas turbine and gasifier, have rapid control loops which are critical for safety reasons as well as for process control. These items would operate automatically and relate to the central supervisory control.

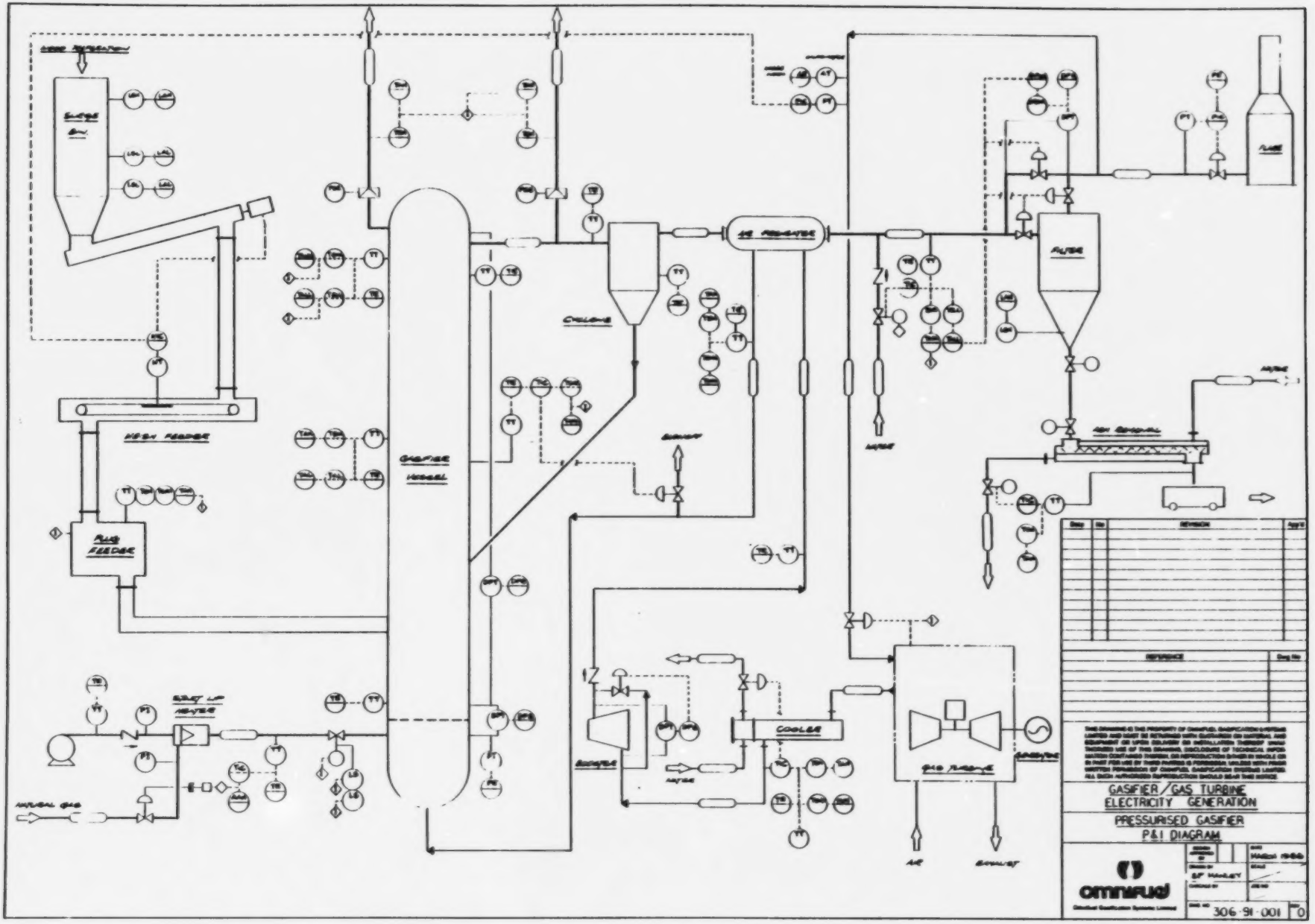

GASIFIER / GAS TURBINE
ELECTRICITY GENERATION
PRESSURISED GASIFIER
P&I DIAGRAM
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FIGURE 5

Operating Modes: There are essentially three modes of operation: startup, operation and shutdown.

1) Startup - To initiate operation, the gasifier must be brought to operating temperature. This will be accomplished by heating the fluidizing air with a startup heater and blower fired by an auxiliary fuel. This would be done at low pressure and would initially take about six hours. Restarts after shutdowns of up to two days would be accomplished within one hour. Shutdowns under 8 hours would require about 15 minutes restart time.

2) Operation - Once the gas is available, the turbine startup sequence can be initiated. This can be either on auxillary fuel or Omnigas. For the demonstration it is recommended that both methods be available, as startup on Omnigas requires whole plant integration and is considerably more complex.

The control systems of the wood preparation, gasifier, gas clean-up, turbine, waste heat boiler are essentially the same as in conventional applications.

3) Shutdown - In the event of equipment failure the system must be capable of shutting itself down in an orderly fashion.

The gas turbine is capable of shutting down extremely rapidly and everything else will follow. The gasifier takes a little longer to shut down and the excess gas will be flared during the shutdown.

# G A S  C L E A N U P

## 4. GAS CLEANUP

### INTRODUCTION

A key component in the commercial application of power generation using a gas turbine operating on combustible gas from Pressurized Fluidized Bed Gasification (PFBG) is that of hot gas cleanup (HGCU). The advantages of using HGCU are that the sensible heat is retained in the product gas (this yields about 5% more power generated) and a liquid waste is not created as in the case of wet scrubbing. For hot cleanup, the temperatures must be kept above 425C to prevent the condensation of tars, the formation of which is already minimized with the use of the fluidized bed type gasifier. Cold cleanup would be the fall back position if hot cleanup does not prove to be technologically or economically viable.

Turbine blades are susceptible to decay by contaminants in the combustion product gas, both in the paticulate and vapour phases. The attack can take the form of erosion by larger particles (> 5μ) or by chemical attack after formation of corrosive metallic eutectic complexes composed mainly of the alkali metals and also Pb and V. These condense out on the blades and in severe cases can necessitate replacement of the blades in a period of days.

At present neither the mechanisms of decay through erosion or corrosion nor turbine life expectancy in this situation have been fully characterized. For erosion considerations, turbine manufacturers give conservative estimates of total dust loading of about 100-200 ppm with particle sizing less than 5μ for reasonable turbine life (25,000 hours). Particles larger than 5μ have enough mass to break through the protective oxide layer covering the blades, nevertheless a few percent of particles 5 - 10μ can be tolerated if the total loading is less than about 30 ppm. It is generally believed that on a large scale, most cyclones cannot achieve the efficiencies needed on fines, and other HGCU systems must be considered for the final stage cleanup.

Much work is presently being performed in the area of HGCU (up to 1,100C, 20 atm.) in relation to power generation through a turbine following coal Pressurized Fluidized Bed Combustion (PFBC). Several types of cleanup systems under development have been tested at the pilot and sub-pilot plant level, but none has large scale commercial experience in this application as yet.

Since no data on HGCU or turbine behaviour after Pressurized Fluidized Bed Gasification of biomass has been identified, this report reviews work performed using PFBC of coal. These results then seen to be conservative when applied to wood, since wood ash is softer than coal ash and contains less alkali metals.

HOT GAS CLEANUP SYSTEMS

HGCU includes high efficiency and electrostatic cyclones, baghouses using ceramic bags to take the higher temperatures (up to 1,100C), rigid ceramic filters, sintered metal filters and granular beds. Electrostatic precipitators and dry sieve plate scrubbers are also being investigated but have not been demonstrated at a similar level.

For PFBG where much of the particulate is smaller than 10μ, cyclones might not achieve the efficiencies required for final stage cleanup, but could be used for a first stage cut. This lowers the size of the more costly final stage and the pressure drops associated with the first stage cyclone(s) are low. Pressure drops across final stage HGCU systems are typically around 3kPa.

Moving granular bed filters are currently being used at temperatures below 400C, and atmospheric pressure, and porous candle filters, both sintered metal and ceramic, are being used at higher temperatures but at low pressures on low radiation waste incinerators in the nuclear industry. Woven and matted ceramic bags, as well as ceramic cross flow filters, are not currently in service in any commercial application.

High temperature electrostatic precipitators (Research Cottrell, Inc.) and dry sieve plate scrubbers (Air Pollution Technology, Inc.) are further from development and have not been investigated in depth for this study.

1. Ceramic Bags

Baghouses using ceramic bags of both the matted and woven form are being developed by both the Accurex Corporation and Westinghouse Electric Corporation using bags manufactured by the 3M Company and ICI Americas, Inc.

The matted media consists of 3 $\mu$ nominal ceramic fibers supported on both sides by either woven ceramic or stainless steel screens, and can be used with a face velocity (air to cloth ratio) up to 10 cm/s. At the high end face velocity, off line cleaning is recommended, while simpler on-line cleaning can be performed with lower face velocities.

The 3M Company produces a woven filter bag made from ceramic yarn. This is used with lower face velocities ( $<$3cm/s) allowing on-line cleaning but necessitating a larger size unit. 3M are currently working on a very fine matted filter.

Cleaning is accomplished by blowing back through the filter with a short duration ( $<$ 1 second) jet pulse of inert gas at a pressure 2-3 times higher than working pressure. The resultant shock breaks loose the cake built up on the bags and the ash falls down into a catch bin from which it is removed. Blinding is less likely to occur with bags than with rigid filters due to the bags' elasticity and thin walls.

Bag filters using a combination of ceramics, glass and metals are currently being developed for temperature applications up to 650C, and should cost considerably less than pure ceramic bags, thus being attractive for use with PFBG.

2. Rigid Ceramic Filters

The two types of rigid ceramic filters are the porous ceramic cross flow filter and the porous ceramic candle.

The former is manufactured by the Coors Procelain Company in conjunction with Westinghouse and is shown in Figure 6. Dirty gas flows though the corrugations parallel to the sealed face, permeates through the surfaces above and below, and exits via the corrugations perpendicular to the inlet flow. Filters currently being tested have rectangular channels. Cleaning is achieved by jet pulse blowback with a pressure about 700 - 1400kPa higher than working pressure. The filter elements could commonly be 15 cm square and 5 cm thick, Their primary advantage is their extremely high surface area-to-volume ratio. This results in a much smaller unit than any other system investigated.

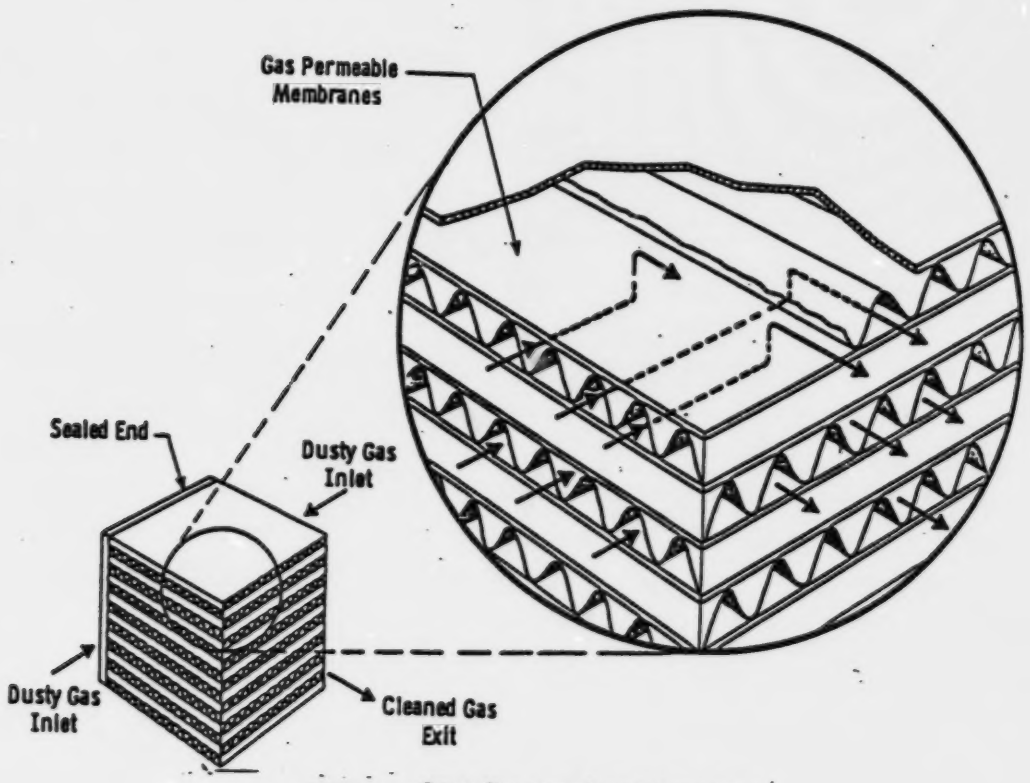



FIGURE 6 —Cross-flow ceramic membrane filter.

(Courtesy of Westinghouse)

Ceramic candles made by Schumacher'sche Fabrik of West Germany have a dense silicon carbide matrix within which are bonded pockets of pure mineral fibers of only 3 $\mu$ in diameter. These candles have relatively thick walls (2 cm.) which tends to compensate for the inherent brittleness of ceramic and its weak mechanical strength. Face velocities are about 10 cm/s and cleaning is achieved again by jet-pulse blowback.

Ceramic has very good high temperature and corrosion stability, but in the above cases of rigid filters, blinding (blocking of pores with particles unable to be blown back) should be considered a possibility. Cost could be higher than other HGCU systems, as could their pressure drop, although not by an order of magnitude.

3. Granular Bed Filters

Shallow beds (10-30 cm.) comprised of pea-sized gravel or other high temperature material can be used for HGCU. The major advantages of these systems are their ruggedness, high face velocities (15-75 cm/s), minimal floor space requirements, and low capital and operating costs, although their efficiencies appear to be slightly less than other systems under investigation. Efficiencies up to 99.98 percent have however have been reported using electrostatic enhancement.

Westinghouse and Ducon, Inc. are jointly testing a fixed granular bed, while the Combustion Power Company Inc. as well as EFB, Inc. have developed an electrified moving bed. Using a 10-20 kv potential difference, nodes of positive and negative charge are established on each granule in the bed, enhancing the collection. One of these designs incorporates a corona discharge through which the gas passes prior to entering the bed. The granules and entrapped particulate continuously move slowly down cross-current to the gas. (See Figure 7). The "dirty" bed is pneumatically lifted to a deentrainment chamber at the top of the filter, the granules recycled and the particulate which is blown loose can be cooled and carried off to a dust receiver.

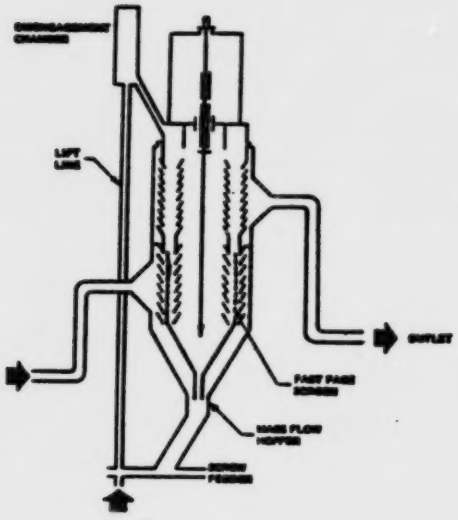


FIGURE 7

ELECTROSTATIC GRANULAR BED FILTER SCHEMATIC

(Courtesy of EPB Inc.)

The electrified beds are very effective at collecting sub-micron particles, but particles greater than 15μ tend to re-entrain and should first be removed with a cyclone or another bed designed for that purpose. The temperature limitation appears to start at about 500-700C, when the bed itself becomes conductive, the potential difference required increases dramatically, and the power cost then becomes significant. A similar effect is observed if the particulate is itself conductive, in which case the dust-to-stone ratio must be low enough to prevent current flowing across the bed. Adding more bed material then necessitates a larger recirculation system and a point is reached at which this becomes no longer economic. Wood gasifier flyash containing high levels of fixed carbon and volatiles has been tested and found to have a resistivity too low to permit electrostatic enhancement of the bed. With process modifications, the fixed carbon in the flyash can be reduced and a level might be reached at which the ash will support the potential difference.

At the higher temperatures (greater than 500C) to compensate for the conductivity of the bed and the observed plugging of louvres with softened particulate in the conventional designs, a modification of the moving bed is being developed. With neither electric enhancement nor louvres or grids, the dirty gas is piped into the middle of the bed and flows up countercurrent to the downward moving granules, as opposed to the earlier crossflow design. (See Figure 8.)

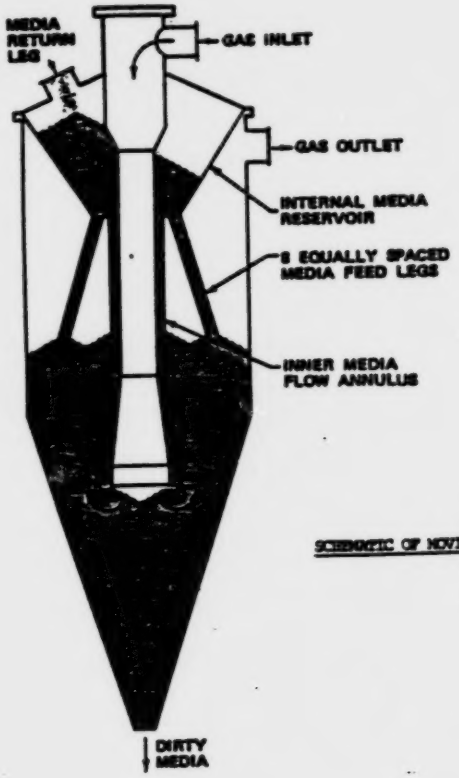



FIGURE 8:
SCHEMATIC OF MOVING SCREENLESS GRANULAR BED FILTER

(Courtesy of Combustion Power Company Inc.)

4. Porous Metal Filters

Sintered metal candles are similar to ceramic candles in shape, size and configuration. They are more stable with respect to mechanical and thermal shock, but are suseptible to both corrosion and blinding, and have a lower temperature limitation. Face velocities are generally in the range of 2 - 6 cm/s and are limited by blinding.

Cleaning is achieved by blowing back for a few seconds at up to 70 percent of the forward flow, or by using a reverse jet pulse method as with ceramic candles.

Two principal suppliers have been identified - Mott Metallurgical Corporation and Pall Trinity Micro Corporation. After characterizing the particulate to be removed, sintered metal filters can be manufactured with suitable porosity. Materials used include stainless steels, Inconel and Hastelloy X.

# G A S  T U R B I N E

5.1 Fuel Feed System

5.2 Combustor Design

5.3 Combustor Types

5.4 Fuel Contaminants and their Effects

5.5 Actual Contaminant Levels

5.6 Turbine Manufacturers

5.7 Criteria for Turbine Selection

5.8 Description of Gas Turbines Selected for this Study

## 5. GAS TURBINE

A gas turbine is a compact efficient means of converting energy in a liquid or gaseous state to mechanical, rotating power. Essentially, there are three basic functions performed by the turbine as follows:

1. Air Compression

A turbine requires large amounts of air for two main reasons; combustion of the fuel and cooling of internal components. In order to use this large quantity of air, it is first necessary to draw the air into the turbine and compress it to a useful working pressure. These functions are performed by the air compressor section. Depending on the type of turbine, this section can range from a single radial stage to a multi-stage axial design as used on newer designs. Compression ratios can vary from 3.5 to 1 to as high as 19 to 1.

This produces a pressurized quantity of air which can then be used for primary combustion, dilution air or for cooling of the equipment.

2. Combustion

The temperature and hence volume of the compressed air is increased by the combustion of fuel. This takes place in one of a number of types of combustors. Some of the air is used for primary combustion of the fuel while the majority is used to quench the products of combustion. This maintains safe temperatures which are within the range of the rotating turbine components.

3. Expansion

The diluted products of combustion are expanded within the power turbine section. This provides power to drive the air compressor section (consuming approximately 2/3 of the power produced) with the remainder available as shaft power. In this particular application, the power is used to produce electricity by means of a generator.

As mentioned previously, some of the compressed air is used for cooling of internal components. This air is normally used within the first or second power turbine stages to help maintain low metal temperatures and promote longer component life.

## TURBINE ADAPTATION

For the most part, turbines have used conventional hydrocarbon fuels as their energy source. Due to their high power to weight ratio, the turbine is extremely well-suited to aviation use where liquid fuels are used due to their high energy density and consistent combustion characteristics.

Stationary turbine applications have a broader energy base ranging from natural gas to the middle distillates as the most popular fuels. Since the most common production turbines are designed to burn these conventional fuels, anything outside this range requires investigation. Although the basic components and functions of the turbine remain the same, some modifications may be required to handle a different fuel.

Compared to conventional fuels, the fuel produced by the gasifier is hot, non-homogenous, and has a relatively low energy density. For these reasons, we must look at a number of specific areas as follows.

### 5.1 Fuel Feed System

The use of this wood-derived gaseous fuel requires careful examination of the design and operation of the fuel system.

It is extremely important to avoid any condensate drop-out as any carry-over into the combustion system could result in mechanical wear or thermal degradation of components, caused by localized hot-spots due to the ignition of volatile condensibles. To prevent this, the fuel should be maintained 20C above the lowest saturation point of any component present.

Earlier experience has shown that fuel produced in the Omnifuel gasifier contains volatile components that will condense below 350C. If it is to be transported or used below this temperature,

it must first be cooled and scrubbed to remove these condensible fractions.

As the gas is produced at 615C, it is more attractive to utilize the sensible heat of the fuel by using it at or above 425C. Use of the fuel at this higher temperature will also enhance the combustibility of the fuel. This will require that the materials used in the fuel system are capable of containing and regulating the gas at these temperatures. The relatively low calorific value requires that a correspondingly high volume of gas must be transported to fulfill the fuel requirements of the turbine. This means, therefore, that fuel supply lines, control valves, and injectors must be adequately sized with cross-sections 10 to 15 times larger than those on turbines burning natural gas.

Although both volume and viscosity tend to increase due to the elevated temperature, the operating pressure will help to reduce the actual volume.

VISCOSITY

At 426C, the viscosity of the gas is about double that at 15C. Final pipe sizing must also take this into account to prevent any excessive pressure losses.

SAFETY

The fuel system must have the necessary integrity and safeguards to ensure against accidental leakage. This is most apparent in the area of valves and flanged joints and these items must be carefully specified to contain the hot fuel gas.

Not just the flammability, but also the toxicity of this gas must be considered. Hydrocarbon detectors must be able to recognize the presence of a leak, especially in any high containment areas where the light, hot gas could rise.

Also possible is a temperature rise by the pressurized gas should it expand because of a leak. Because of negative Joule-Thomson effects, and the lack of work being done (isenthalpic expansion), continued leakage could result in attaining auto-ignition temperature and subsequent combustion.

## 5.2 Combustor Design

The particular type of combustor design is one of the most critical aspects in the successful operation of the turbine. In burning a low heating value fuel, many different aspects of the combustion process must be considered.

A number of these factors and their impact on the turbine design and operation are discussed in this section.

WOBBE INDEX

Gas handling capacity of the combustor nozzles can be determined using the Wobbe Index. This is referred to as the gas index by some manufacturers when the lower heating value is used instead of the higher heating value, in other words;

$$\text{Wobbe Index} = \frac{HHV}{\sqrt{S.G.}} \qquad \text{Gas Index} = \frac{LHV}{\sqrt{S.G.}}$$

Where: HHV = Higher Heating Value, BTU/scf

LHV = lower heating value, BTU/scf

$$S.G. = \frac{\text{density of gas (STP)}}{\text{density of air (STP)}}$$

$$= \frac{\text{mol. weight of fuel gas}}{\text{mol. weight of dry air}}$$

The index is very useful in determining interchangeability between fuels. Although the calorific value of the fuel is important in determining combustion characteristics, specific gravity is also important since the metering and flow characteristics of nozzles and valves are related to the volumetric flow. Fuels having widely different calorific values could behave similarly given a corresponding and opposite variation in the specific gravity.

As many turbine combustion systems were designed for high heating value fuels such as natural gas, compatability of system handling capability using the wood-derived gas can be checked by using one of the above indices.

Standard turbine combustion equipment is capable of handling gas with a Wobbe index of about 950. Variations within plus or minus 15% are attained by making minor adjustments to the throttle valve in the fuel control system.

For lower values, more serious modifications are required to accommodate the greater mass flow of the low heating value fuel. Nozzles and inlet lines must be enlarged to allow sufficient fuel flow while keeping pressure losses within reasonable limits.

Based on a calorific value of 5.24 MJ/Nm3 (133 BTU/scf) and specific gravity of .94, the Wobbe Index for the fuel used in this report is about 140.

Use of this gas at elevated temperatures could further compound delivery problems although this will be compensated by the fact that it will be pressurized.

## COMBUSTION TEMPERATURE

The low heating value of the fuel gas and the large quantity of inert gases (water vapour and nitrogen) that comprise part of the fuel gas, result in relatively low adiabatic combustion temperatures. The adiabatic flame temperatures with stochiometric air are in the region of 1650C vs. 1900C for natural gas. Since large amounts of excess air are added in order to dilute the temperature to between 950 - 1100C before entering the turbine expander, this should not be a problem as far as the expander is concerned, but it does need to be considered in the combustor where the flame stability is of major concern.

## COMBUSTION AIR REQUIREMENTS

The volume of the fuel gas and its products of combustion can vary greatly from that of conventional fuels.

The stochiometric air requirements and products of combustion are approximately as follows:

TABLE 1

Comparison of Fuel Combustion Volumes

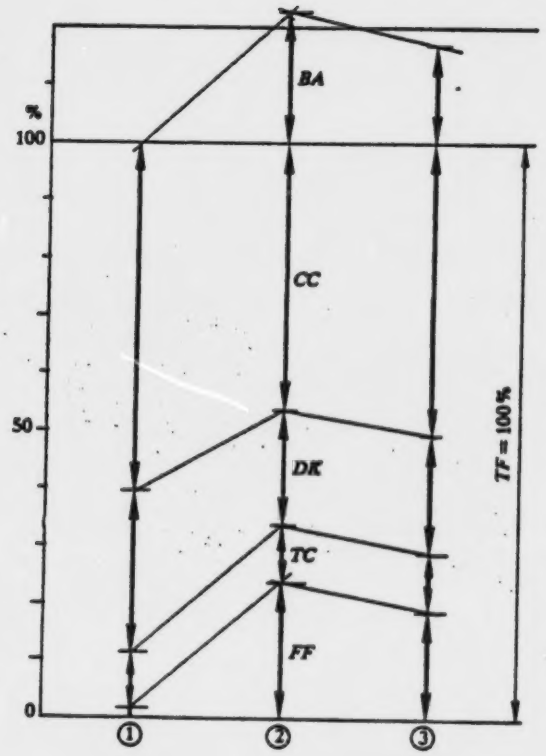
| | Low Heating Value Fuel | Methane |
|---|---|---|
| Lower Heating Value (MJ/Nm3) | 5.24 | 36.0 |
| Air Required (cu.m/m3 gas) | 1.17 | 9.53 |
| Air Required per GJ Heat Input (Nm3) | 235 | 280 |
| Combustion Products per GJ, Heat Input (Nm3) | 420 | 310 |

The design of the primary combustion zone, therefore, must take this increased volume into account. Even allowing for a lower adiabatic flame temperature, this increased volume will be substantial. Figure 9 is a comparison of gas volumes in the gas turbine cycle during combustion of methane, blast furnace gas and gas produced by oil gasification.

Fortunately, the disparity in volumes is reduced as dilution air is added to reduce the turbine inlet temperature. In our study, excess air varies between 230% and 400% depending on the turbine model. This air is normally added in the combustion chamber to assist in cooling of the liner walls and provide maximum time for mixing of the cooler gases. In most instances, the turbine expander cannot take the additional volumes produced from low heating value gas (BA in Figure 9) and air must be bled from the turbine and either used in the gasifier or discharged to atmosphere. In this case, the dimensions of the latter sections of the combustor should not require any substantial revisions.

## COMBUSTION STABILITY

The low adiabatic flame temperature, lower flame speeds, and inherent variability of the gas due to the non-homogeneity of the wood feedstock make the combustion less stable than natural gas or oil. In addition to this, there are small particles of char present in the gas.




Comparison of gas flow rates in gas-turbine cycle during combustion of methane, blast-furnace gas, and gas produced by oil gasification

*FF* = Fuel
*TC* = Turbine cooling air
*DK* = Combustion air at $\lambda = 1$
*CC* = Cooling and mixing air in combustion chamber
*BA* = Excess air
*TF* = Exhaust gas from turbine
① = Methane
② = Blast-furnace gas
③ = Gas produced by oil gasification

FIGURE 9

Courtesy of Brown Boveri

To ensure stable and complete combustion, a combustor with a long residence time is recommended. (See Figure 10) This not only allows for greater combustion of char particles having long residence times, but also effectively reduces emissions of carbon monoxide (CO) and unburned hydrocarbons (UHC).

As residence time is a function of gas velocity and distance travelled, a long combustor having a relatively large cross-sectional area would provide the longest residence time.

RATIO OF FLAMMABILITY

The ratio between the upper and lower limits of flammability is critical as an indicator of the turbine operating flexibility. Since the turbine generator set operates at a constant speed, the volume of air entering the turbine will be relatively constant.

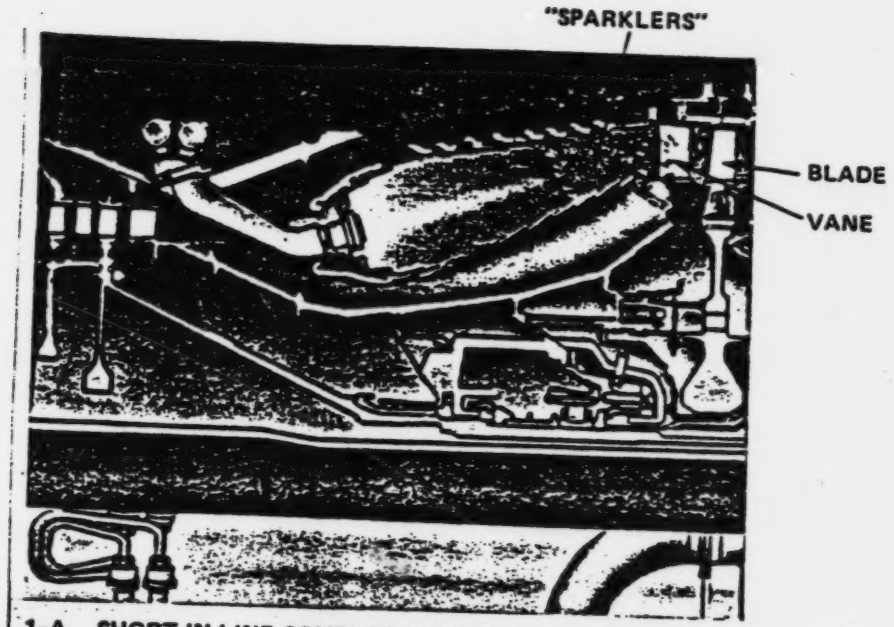
In order to handle fluctuating loads, the amount of fuel used will be adjusted by the throttle control valve. However, there is no such adjustment on the combustion air volume, so the fuel must be capable of burning at different levels of excess air.

In order to achieve a proper turn-down in performance, the fuel should have a minimum ratio of flammability of 2.8 to 1.0. Below this, modifications will likely be required to the fuel control or combustion systems.

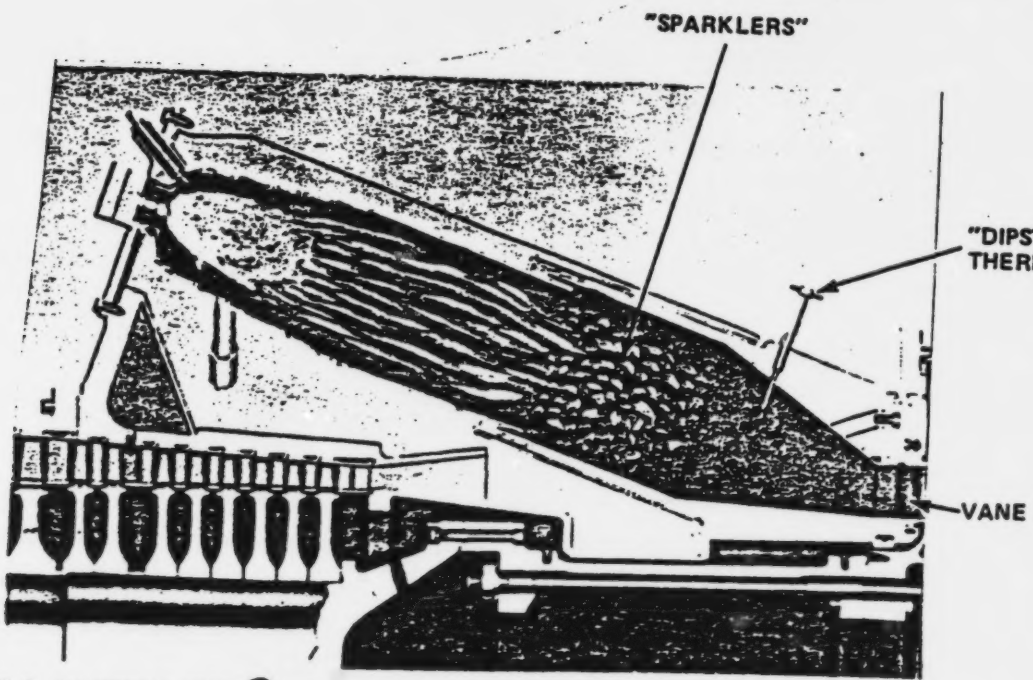
Not only must the turbine be able to run at part load without fear of a flame-out, but starting and stopping operation can also necessitate substantial flame retention capabilities. For these reasons, the turbine system as envisaged here would have dual fuel capacity with the standby fuel system capable of providing up to 100% operation.

OXIDES OF NITROGEN

The high nitrogen content of the low heating value fuel could result in undersirable emissions of oxides of nitrogen (NOx). Since no after-burning or catalytic device is used on the turbines, one alternative method of NOx control involves




1-A SHORT IN-LINE COMBUSTOR WITH VERY SHORT TRANSITION "SPARKLERS" IMPINGE ON TURBINE VANES AND BLADES




1-B CANTED LONG (W) COMBUSTOR BASKETS TRANSITION ALLOWS "SPARKLERS" TO COOL TO GAS TEMPERATURE BEFORE THEY ENTER TURBINE VANES

FIGURE 10- COMBUSTOR/TRANSITION SECTION OF GAS TURBINES OF SIMILAR SCALE

injecting water or steam into the primary zone of the combustor. Basically there are two combustion processes which cause the formation of NOx.

The first is the oxidation of atomic fuel bound nitrogen. This should not present a serious problem due to the fact that wood contains very little fuel bound nitrogen.

The second source is the thermal fixation of molecular nitrogen at higher combustion temperatures. Since this process is a function of high temperature and residence time, we can examine factors affecting this process. The relatively low temperature at which the wood is gasified should also help limit the production of NOx.

To reduce carbon monoxide and unburned hydrocarbons, adequate residence time is essential. Since this is a function of the hardware, we will assume this is fixed by the turbine manufacture's design. Although the actual fuel mass vis a vis natural gas is greater for the fuel in question, the total mass through the combustor should remain approximately the same by reducing the amount of dilution air. This will tend to maintain the same velocities and, therefore, residence time.

Control of NOx can then be achieved by reducing the temperature within the primary combustion zone by adding water or steam. However, the low heating value gas has a lower flame temperature than oil or natural gas and hence the production of NOx in the combustor would be expected to be lower. The gas inerts (water, nitrogen, carbon dioxide) have the same effect as steam injection, used to control NOx with conventional fuels.

## 5.3 Combustor Types

In normal practice, there are three basic types of turbine combustors; annular, can-annular, and single can.

### 1) Annular

The annular combustor is essentially an enclosed ring shaped chamber surrounding the turbine rotor. (Figure 11) There are no separate interior chambers and pressure losses are minimal.

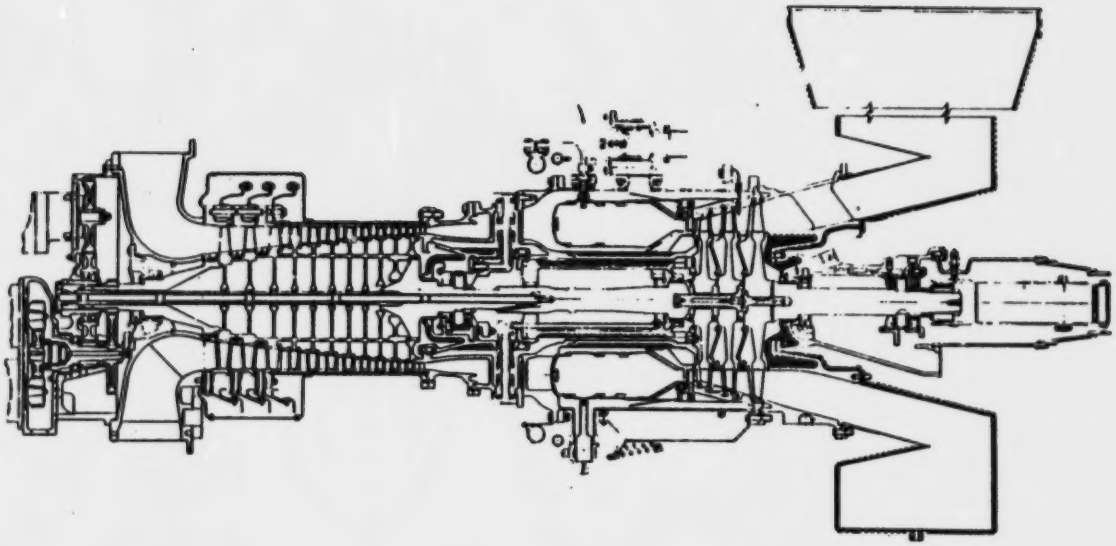


FIGURE 11 - Cross Section of Gas Turbine
with Annular Combustor

Because it surrounds the engine, however, combustor replacement involves complete turbine disassembly. This could prove extremely expensive and burdensome when burning alternative fuels which may necessitate combustor maintainence.

In addition to this, it could be difficult to introduce the volume of low heating value fuel to the annular combustor without extensive modifications to the combustor and turbine casing.

Maintaining a stable flame front in an annular combustor may also be more difficult than with a can-type combustor.

2) Can-Annular

The can-annular arrangement consists of individual combustor cans arranged symmetrically about the central shaft. (Figure 12) Because the diameter of the combustor is usually rather small (as opposed to single can combustors) thin film skin cooling is normally used to cool the combustor liner.

Parasitic pressure losses are low and the multiple cans can help prevent the turbine wheel from seeing extreme temperature changes. In other words, the greater the number of cans, the more uniform the temperature of the combustion products will be in a 360 degree circle.

Since the individual cans are normally removable, maintenance and replacement can be greatly simplified.

The can-annular arrangement can also be subdivided into two categories, the in-line type and the canted type. As the name implies, the in-line type is parallel to the rotor shaft and is nestled between the last stage of the compressor and the first stage turbine nozzles. To prevent torsional or dynamic stability problems in the shaft, this distance must be relatively short. This in turn necessitates a short combustor with a corresponding short residence time. This can allow glowing carbon particles to

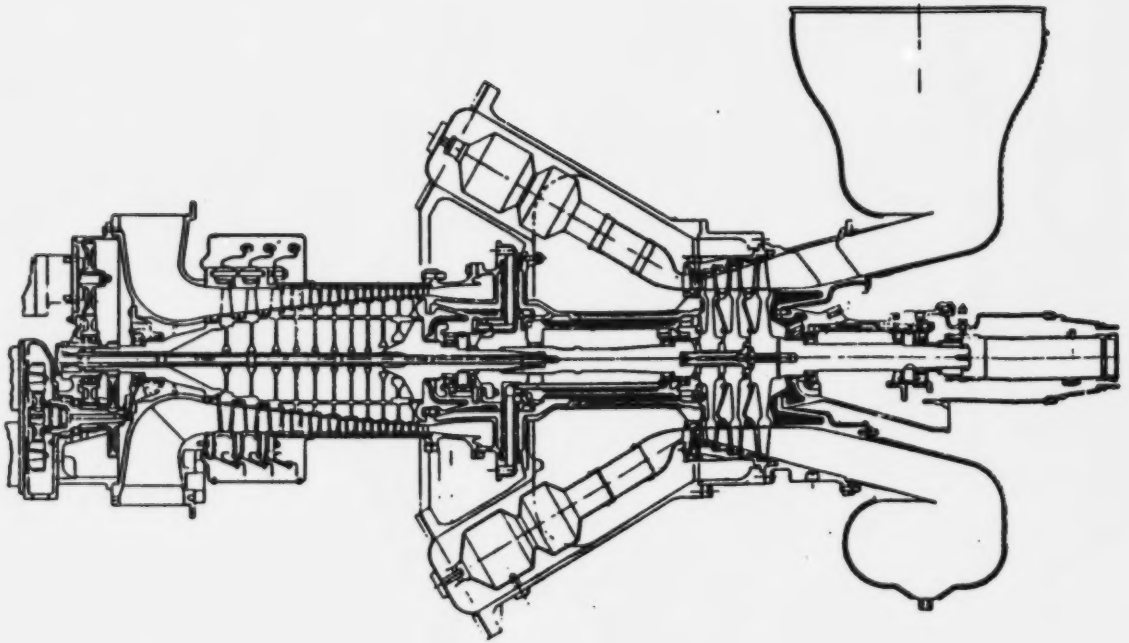


FIGURE 12 - Cross Section of Gas Turbine
with Can-Annular Combustor

remain alight as they impact the turbine nozzles or wheels and cause a localized hot spot with a reducing atmosphere. This penetration of the protective oxide layer can cause serious corrosion problems.

In order to alleviate this situation, a canted arrangement can be used. This allows a longer residence time by extending the combustor on an angle beyond the air compressor section.

Pressure loss is somewhat greater than an in-line combustor due to the reverse air-flow and longer path through the combustor. This negative feature, however, is offset by the increased benefits of the residence time. The external orientation of the cans will also permit the use of large fuel lines to feed the gas to the combustor nozzles.

3) Single Can

The third type is the external single-can combustor. The remote placement and large dimensions of this type of combustor allow great flexibility when burning various fuels. (See Figure 13)

Residence time can be extremely long for efficient and thorough combustion. Pressure loss is low, but some losses can occur due to the elbows and transitions used to remove the compressed air and then reintroduce the combustion products.

## 5.4 Fuel Contaminants and their Effects

The modern gas turbine is a very sophisticated piece of equipment. Notwithstanding the relative simplicity of the Brayton cycle itself, the demands of fuel efficiency and performance lead to the use of materials and configurations which stress these components to the upper limits.

Because of this low margin for wear or fatigue that is inherent in a gas turbine, any introduction of contaminants that affect these items must be carefully examined. This is especially critical along the hot gas path where elevated temperatures can accentuate the deleterious effects.

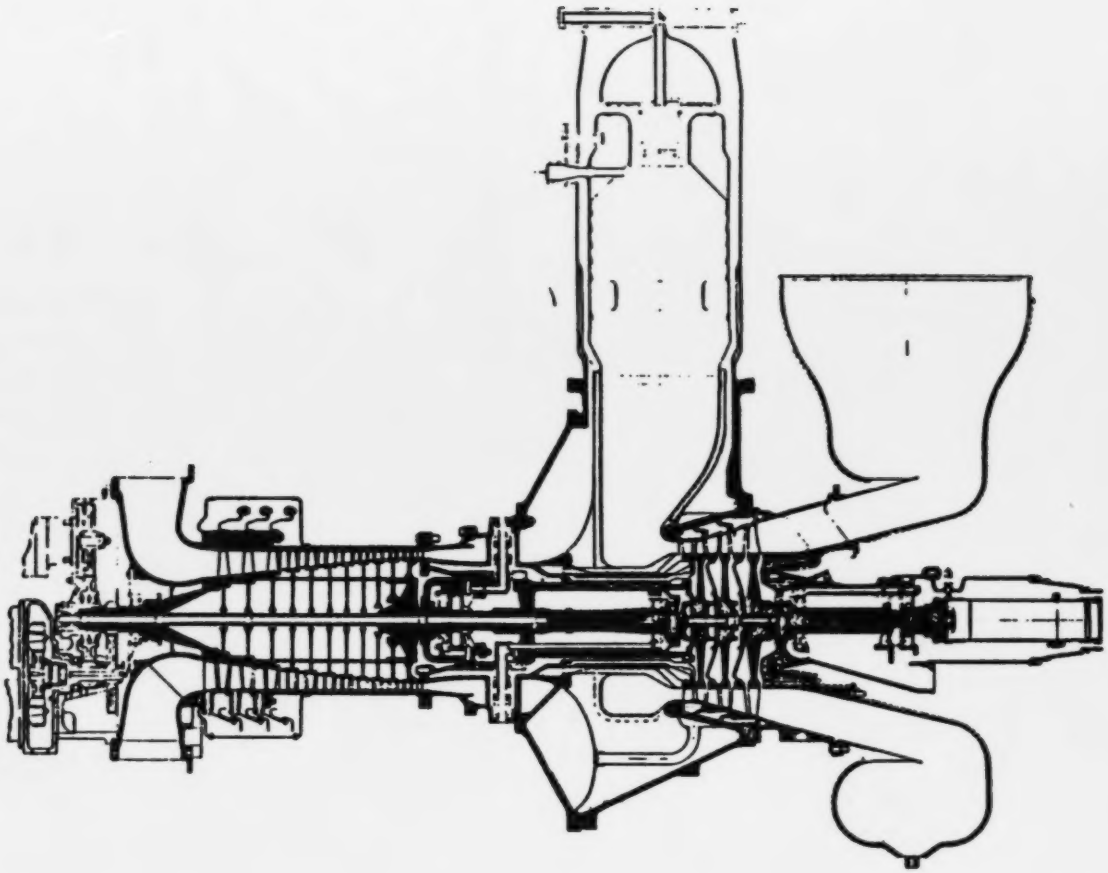


FIGURE 13 - Cross Section of Gas Turbine with Single-Can Combustor

For this reason, we will examine possible contaminants from the fuel gas and their respective effect on the turbine life.

We should bear in mind that there are really no absolutes in contaminant levels. Even a small amount of corrosive or erosive material will cause some damage. For this reason manufacturers usually specify maximum levels that will give acceptable life cycles. These are the limits used in the following discussion.

VANADIUM

Vanadium can combine with other elements in the gas path to form compounds such as vanadium pentoxide (V205) and sodium metavanadate (NaV03) at temperatures as low as 690C. Low melting eutectics comprised of these materials can melt at 580C. These compounds are highly corrosive and can severely damage the power turbine blades and nozzles.

To reduce the effects of vanadium, magnesium can be added at the point of combustion. This forms magnesium vanadates which have a high melting point and will generally pass through the turbine.

Silicon can also be added to give a friable compound which is more easily removed during cleaning. Coating of the blades with a protective layer is also effective and is highly recommended to prevent corrosion.

SODIUM AND POTASSIUM

These alkali metals can combine with vanadium or sulphur to produce compounds which can occur over a wide temperature range such as found in normal turbine operation.

These alkali sulphates, sodium vanadates and similar compounds have proved to be highly corrosive. Sodium sulphate (Na2 S04), for instance, has a melting point of 874C (1,605F.). Where parts of the turbine exceed this temperature, the molten sodium can contact these components and remain in a molten state. This liquid compound can then act as a "flux" and form a reducing atmosphere which removes the outer oxide layer protecting the metal underneath.

If practical, water washing of the fuel can significantly reduce the level of these contaminants. The use of corrosion resistant high-temperature materials and blade coating is also recommended.

LEAD

Lead by itself is corrosive. It can also combine with vanadium, sodium or potassium to form more highly corrosive compounds.

Where magnesium is used to control vanadium, lead can contaminate the magnesium and render it ineffective.

CALCIUM

Calcium can be useful by inhibiting the corrosive action of vanadium. On its own, deposition can occur which fouls the aerodynamics of the blades and reduces power and efficiency.

If calcium is present in the fuel, periodic water washing of the turbine blades is recommended. Water washing of the fuel can also help.

SULPHUR

Sulphur in the fuel can combine with oxygen and sodium or potassium compounds to form sulphates, pyrosulphates and sodium or potassium iron trisulphate. These can melt at operating temperatures and cause severe corrosion.

Hydrogen sulphide and carbonyl sulphide may also be present in the fuel. These compounds could be damaging to the fuel supply system as well as downstream heat recovery equipment.

Fuel based sulphur must also be controlled to limit the formation of sulphur oxides.

Since the amount of these compounds in the exhaust is directly related to the amount of sulphur in the fuel, control is also required to meet emission standards. For a combined heat and power installation, low sulphur levels in the exhaust are

required to allow a minimum stack temperature for optimum heat recovery.

IRON

It has been found in an application burning blast furnace gas, that iron or iron oxide can have an impact on carbon deposition where carbon monoxide is present. (1)

The gas we are studying has about 12% carbon monoxide (CO). The CO is unstable between 300 and 600C and when it comes into contact with iron or iron oxide, the following reaction can take place:

$$2\ CO + \text{catalyst} = CO_2 + C$$

This can cause a deposition of carbon which could occur anywhere in a system where component temperatures are between 300 and 600C.

In the system under review, this could possibly occur in the heat exchanger or gas cleaning filter. Particular attention should be paid to blinding of the filter elements and it may be desirable to run the filter at the higher gasifier exit temperature.

SILICA

The silica present can play two roles as a gas contaminant.

1. As a particulate, its size and quantity must be limited to prevent erosion of the turbine components.

2. As a reactant, the presence of silica can be beneficial in preventing hot path corrosion. As silicon oxide, it can react with other contaminants present to raise their melting point and help prevent deposition of these compounds in a liquid state.

MAGNESIUM

Magnesium oxide can be very useful in raising the fusion temperature of certain cohtaminant compounds. For example, three

(1) Brown Boveri Review, January, 1977, Volume 64.

parts magnesium to one part vanadium will increase the melting point above the operating temperature of the hot turbine components.

This can lead to deposition but the magnesium compounds have been found to be very friable and easily removed through regular water washing of the turbine blades.

AMMONIA

Some of the ammonia in the fuel is converted directly to NOx and cannot be reduced by either water or steam injection.

Control of this component in the fuel gas is thus required for environmental protection.

PARTICULATE MATTER

Particulate matter in the fuel gas must be controlled to protect the turbine from erosion, corrosion and scale deposition.

Some features of the particulate which would determine its effects are hardness, shape, surface roughness, size and quantity. The first three items are really a function of the particulate composition and the processes which create the particles. These variables are fixed by the fuel gas production and are basically outside of any control.

The latter two qualities, size and quantity, can be controlled through cleaning procedures as outlined elsewhere in this report.

Opinions expressed by the turbine manufacturers vary on the maximum allowable size of particulates. One manufacturer feels that particles must be under 20 microns in size. Another feels 6 microns is a maximum with the average size being 3 microns or less.

In general, the smaller the particle, the less likely it is to penetrate the boundary layer of the blades at damaging velocities. Particles under 2 microns will likely have no erosive effects.

Particulate limits in the fuel are set to give an acceptable time between overhauls. These limits are normally based on a fuel such as natural gas and are expressed as a function of fuel flow. When burning low heating value fuel, it is important to recognize the increased flow rates and to reduce the limits accordingly. Total particulate throughput must not exceed the limits for standard fuel on the basis of equivalent mass or volume per unit energy input.

5.5 Actual Fuel Contaminants

Tables 2 and 3 show typical compositions of wood and peat ash used as feedstock to the gasifier. Levels of ash are 1.0% for the wood and 5.0% for the peat. It is assumed that all of the ash carries over into the gasifier stream. No carryover of the bed material is included as this amount is minimal in comparison to the feedstock ash.

The composition is expressed as a percentage of the ash. Using these figures, it is possible to determine the contaminant levels in the gas stream.

In actual operation, we expect the primary cyclone to be 70% efficient and the secondary filter gas clean-up system to be at least 99.9% efficient in removing particulate matter. The final levels are shown compared to typical maximum levels recommended for a gas turbine.

In most instances, contaminant levels are well within the limits set by the turbine manufacturers, especially with wood as the fuel. In the case of wood, only total sodium and potassium exceeds the guidelines and then only by .02 wppm.

With peat as a feedstock, iron and silica carryover exceed the limits. If we assume that these items are more injurious from an erosive as opposed to corrosive sense, then their impact can be reduced if small particle sizes can be maintained. This is quite probable since the gas clean-up equipment is more effective on the larger particle sizes.

TABLE 2

TYPICAL WOOD FUEL

| Contaminant | % of ash by weight | Concentration in gas from gasifier ppm | Concentration after gas cleaning eqpt. ppm | Typical Gas Turbine Requirements wppm of LHV gas |
|---|---|---|---|---|
| Vanadium | Nil | | | |
| Sodium | 2.6 | 86.1 | .026 | .1 |
| Potassium | 14.7 | 487.4 | .15 | |
| Total Na & K | 17.3 | 573.5 | .17 | .15 |
| Lead | Nil | | | .15 |
| Calcium | 26.7 | 885.5 | .26 | 1.0 |
| Sulphur | 2.2 | 72.9 | .02 | 1000 |
| Ammonia | Nil | | | Site Specific |
| Magnesium | 4.7 | 155.8 | .047 | .15 |
| Iron | 1.05 | 34.8 | .010 | .15 |
| Silica | 1.1 | 36.5 | .011 | 1.5 |
| Phosphorous | 5.0 | 165.8 | .050 | N/A |
| Chlorine | 0.6 | 20.0 | .006 | .15 |
| Oxides | 23.0 | 762.6 | .228 | N/A |
| Miscellaneous | 2.0 | | | |
| Total Particulates | 100.0 | 3316 | 1.00 | 15 |

**Notes**

1. Contaminants are present in the gas stream as particulate matter.
2. One kg of wood (dry basis) produces 2.8 m3 of gas at 5.24 MJ/Nm3.
3. Typical ash content of wood is 1.0%.
4. Gas clean-up equipment efficencies: primary (cyclone) 70%, secondary (filter) 99.9%.
5. Assume 100% carryover of ash from gasifier bed.

TABLE 3

TYPICAL PEAT FUEL

| Contaminant | % of ash by weight | Concentration in gas from gasifier wppm | Concentration after gas cleaning eqpt. wppm | Typical Gas Turbine Requirements for low HV fuel, wppm |
|---|---|---|---|---|
| Vanadium | .1 wppm | .0015 | $450 \times 10^{-9}$ | .1 |
| Sodium | .1 wppm | .0015 | $450 \times 10^{-9}$ | -- |
| Potassium | 112 wppm | 1.70 | $510 \times 10^{-6}$ | -- |
| Total Na & K | 112 wppm | 1.70 | $510 \times 10^{-6}$ | .15 |
| Calcium | .49% | 74.2 | .02 | 1.0 |
| Sulphur | -- | -- | -- | 1000 |
| Iron | 2.7% | 409 | .12 | .15 |
| Silica | 45.9% | 6954 | 2.1 | 1.5 |
| Nickel | .1 wppm | .0015 | $450 \times 10^{-9}$ | .15 (Assume the same as heavy metals.) |
| Total Particulate | 100% | 15,151 | 4.55 | 15 |

Notes

1. Contaminants are present in the gas stream as particulate matter.

2. One kg of peat (dry basis) produces 3.1 m3 scf of gas at 5.24 MJ/Nm3.

3. Typical ash content of peat is 5%.

4. Gas clean-up equipment efficiencies: primary (cyclone) 70%, secondary (filter) 99.9%.

5. Assume 100% carryover of ash from gasifier bed.

## 5.6 Turbine Manufacturers

The study began with an initial contact of turbine vendors. Manufacturers around the world were given typical gas conditions and asked to respond with information on their ability and experience in this field.

The manufacturers contacted are listed in Appendix "A". Individual companies interested in this particular application are described below:

### BROWN, BOVERI AND COMPANY

BBC has considerable experience in the use of lower heating value fuel, mostly blast furnace gas. These actual installations have made them aware of many potential problems in the use of alternative fuels.

Their single-can combustion system appears very well suited for the combustion of low heating value gas and their continuing research and development efforts allow them to address questions related to the use of these gases.

### DETROIT DIESEL ALLISON

The DDA turbines are basically aero-derivative engines with many years service in various applications.

The current combustor design is best suited for conventional fuels and modifications will likely include an external combustor. Development work is currently underway on a unit which will burn sawdust and could be adapted for low heating value gas.

### FERN ENGINEERING (Thomassen, U.S.)

Fern does considerable work in modifying and servicing turbines manufactured by others. As the North American representative for Thomassen Holland, they are also involved in development work for the TF 10.

This model is currently in the engineering phase. Calculations show it will have an ISO rating of about 7,700kW and a simple cycle thermal efficiency of 43 to 44%. Compression ratio would be 9:1 and the turbine inlet temperature 1,115C.

The engine is designed to accept an optional combustion liner which is longer than the normal one. This would be used for very low heating value gases although recent testing has shown the regular combustor capable of operating with gas as low as 6 MJ/Nm3.

Introduction of the TF 10 is at least two years away but a low heating value model could be supplied at that time as well.

GENERAL ELECTRIC

General Electric is devoting considerable attention to turbine/coal gasification technology. Based on their experience to date, they believe that the gas in question could be burned in their equipment with the following reservations:

a) The flammability ratio of the gas might not allow lighting-off of the turbine. Pilot fuel would probably be required for start-up.

b) The larger fuel flow would require larger fuel gas piping, control valves, fuel nozzles and a special combustion wrapper.

As for the quality of the gas, total solids would be limited to 30 wppm with no particle larger than 10 microns. Gas must have at least 30C superheat to prevent condensation. Compared to standard natural gas, they would expect a 5% increase in power output and a 1.5% decrease in heat rate.

Their on-going efforts involve continued investigation of the effects of lower heating value gas on component life. They believe that 25,000 hours blade life is possible with particulate loading as high as 23 wppm (less than 0.1% larger than 10 microns). These are levels that could possibly be achieved using cyclones alone for gas clean-up.

MITSUI

Mitsui has a number of turbine models which have external combustors suitable for this application.

They have considerable experience burning blast furnace gas and cannot foresee any major problems in using the gas in question.

NATCO (Kongsberg)

NATCO responded favourably to our request for information on turbines suitable for using low heating value gas. Their KG-2 model has many features which are attractive to this application and their support facilities were helpful in presenting performance data using this fuel.

PRATT AND WHITNEY

Pratt and Whitney turbines are designed and used mostly on liquid fuels. Because of this, they do not really have any equipment to offer at this time.

They did say, however, that they would be interested in developing such equipment given the proper market conditions and incentives.

ROLLS-ROYCE LTD.

Rolls-Royce has been involved in considerable efforts recently in adapting their turbines to use various fuels. Together with Foster-Wheeler, they are offering a combined cycle power plant using a coal gasifier.

At present they believe their engines are capable of operating on gas as low as 8 MJ/Nm3. 4 MJ/Nm3 gas would necessitate major changes in the combustion system.

WESTINGHOUSE

Westinghouse is heavily involved in the development of their turbines to burn low heating value fuel. Some of their earliest

efforts date back to 1953 when a combustor was developed for 3.5 MJ/Nm3 blast furnace gas.

At present their efforts include the determination of contaminant and particulate levels when burning low heating value fuel. They feel that recent developments in metallurgy and turbine technology will allow uprating of both power levels and component life when burning alternative fuels.

They appear to be definitely committed to the use of these fuels and were very helpful in submitting comprehensive performance and technical data.

## 5.7 Criteria for Turbine Selection

From the responses received, it was necessary to select a number of turbines to receive detailed study. In order to quantify the appropriate features, the following criteria were established and the equipment ranked on a scale from 0 (poor) to 3 (good). Those units having the highest scores were selected.

- experience

- committment to further development

- fuel consumption/efficiency

  0 less than 20% simple cycle thermal efficiency
  1 20 - 25%
  2 25 - 30%
  3 greater than 30%

- contaminant levels

- transient response, i.e. resistance to changes in heating value of the fuel

- combustor design

  1 for annular
  2 for can-annular
  3 for external can

- degree of modifications required

  1 considerable
  2 average
  3 very little

- adaptability to pressurized fluid-bed gasifier

  0 greater than 2500 kPa (abs) inlet gas pressure
  1 2,000 - 2,500
  2 1,500 - 2,000
  3 less than 1,500

- recoverable exhaust heat

  0 less than 450 degrees C exhaust temperature
  1 450 - 500
  2 500 - 550
  3 greater than 500

- Fuel temperature

  0 less than 426 degrees C
  3 greater than 426 degrees C

## 5.8 Description of Gas Turbines Selected for this Study

In order to weigh the merits and economics of a combined gasifier/turbine system, a number of different turbine sets were investigated. The final four that were selected are outlined below.

NATCO KG-2

The KG-2 is nominally rated at 1410 kW ISO (15 degrees C, sea level) with no inlet or exhaust losses. It is a single shaft turbine with a single-stage radial compressor, single-stage radial turbine and single tangential combustor. The compressor and the power turbine are mounted back-to-back and overhung on tilting pad journal bearings in the turbine casing.

Power is directed through an integral parallel shaft reduction gear unit to give an 1800 rpm output speed. The turbine and generator are both mounted on a single-skid, complete with lube system and electrical connections. Combined weight of approximately 15,000 kgs.

WESTINGHOUSE 191

The W191 was first introduced in 1965. Since that time, it has continued to evolve with various upratings to its current output level of 20 MW ISO.

The turbine is a simple cycle, single shaft, cold end drive design. In other words, the prime power is taken from the air inlet end of the engine. The compressor section consists of 15 axial stages and has a combined compresssion ratio of 7.6:1. The casing is of a single horizontally split design. Multiple can-type combustors are canted between the compressor and turbine.

The turbine section consists of 5 axial stages with a maximum inlet temperature of 845C.

The engine is mounted on a bedplate along with the integral lubricating oil reservoir, lube oil pumps, auxiliary gear unit, turning gear and starting equipment. An auxiliary skid contains the fuel controls, control air compressors, auxiliary motor controls, pressure switch and gauge panel.

Altogether the package measures 37 m long, 5 m wide and 4 m high and weighs 145,000 kg.

BBC TYPE 9

The Type 9 is a single-shaft, cold end drive turbine with a single, externally mounted can-type combustor. ISO rating is 34.5 MW.

The compressor has 15 axial stages with a total pressure ratio of 8.85:1. There are four turbine stages. The single combustor is complete with nozzle, swirl vanes and ignition torch.

The generator is air cooled and directly mounted on concrete foundations. The gear assembly is separate as is the control room which includes the control system, electrical auxiliaries and switchgear.

The engines themselves measure 10 m long, 3.2 m wide and 7.6 m high. Weight is about 79,000 kg.

WESTINGHOUSE 251

The Westinghouse 251 is a single-shaft, cold end drive turbine set. ISO rating is 41.3 MW, compression ratio is 14:1 and

turbine inlet temperature is 1085C. The engine complete with the reduction gear assembly, associated piping and wiring is mounted on a steel base incorporating an integral lube oil reservoir.

Generator and exciter are separately mounted as is the starting package and the electrical/control package. The starting system includes a diesel engine, gears and coupling while the electrical/control panel includes the required motor control center and protection equipment.

Overall dimensions are approximately 34 m long, 10 m wide and 8 m high.

# A U X I L I A R Y  E Q U I P M E N T

6.1 Waste Heat Recovery

6.2 Air Inlet System

6.3 Switchgear

## 6. AUXILIARY EQUIPMENT

### 6.1 Waste Heat Recovery

The turbines we have examined vary greatly in their single cycle thermal efficiency. This can range from a low of 16.9% for the Natco KG-2 to slightly over 30% for the Westinghouse 251 and new developments promise to increase these to 40% in the near future.

The newer, high compression units usually have higher efficiencies. Due to improved metallurgy and features such as blade cooling, they run with higher turbine inlet temperatures. This can result in relatively hot exhaust temperatures that are well suited for heat recovery.

Based on a stack temperature of 165C, the turbines have a maximum available exhaust energy as outlined in Table 4.

By recovering energy in the turbine exhaust, the increase in efficiency is dramatic, going from an average 26.0% to 75.6% (with 176C stack). It is attractive, therefore, to investigate means of using this energy. This may be achieved in a number of ways, all of which have associated features as follows.

1. Direct Contract Drying - This is the simplest means of utilizing the exhaust heat. A rotary dryer would have a minimal back pressure but turbine performance could be impeded by stack gas clean up equipment. For the Westinghouse 191, this loss is about 200 kW per kPa back pressure.

Part of the exhaust heat could be used for drying the wood feed and is described elsewhere in this report.

2. Fluid Heating - Fluid heating systems are cheaper than steam since only one phase is involved. Water, water/glycol mixture or organic heat transfer fluids can be used depending on operating temperatures. This method would be most applicable to a system requiring the transfer of low grade heat such as process or space heating.

TABLE 4

| EXHAUST HEAT AVAILABLE | | | | |
|---|---|---|---|---|
| | NATCO KG2 | WESTINGHOUSE 191D | BBC Type 9 | WESTINGHOUSE 251 |
| Fuel Input (MW) | 8.5 | 74 | 120 | 134 |
| Electrical Output (MW) | 1.43 | 21.70 | 32.80 | 41.28 |
| Efficiency (%) | 16.9 | 29.4 | 27.3 | 30.7 |
| Exhaust Mass Flow (kg/sec) | 14.1 | 131 | 155 | 159 |
| Exhaust Temperature (degrees C) | 487 | 452 | 520 | 518 |
| Energy Available (MW, 176C) | 4.7 | 37.5 | 57.4 | 58.6 |
| Overall Efficiency (%) | 72.7 | 80.2 | 75.2 | 74.3 |

3. Steam - The use of these turbine exhaust gases for the production of steam is perhaps the most common use. Not only can steam be used directly in a process, but additional electrical power could be produced by using the steam to drive a steam turbine generator set. Use of a back pressure or extraction type turbine would produce a combination of steam generated electricity and process or space heating steam.

4. Other Applications - One other possible use of the turbine exhaust would be the direct heating of a greenhouse following dilution with ambient air. This is now being practised at one location in Ontario where certain produce thrives in the CO2 enriched atmosphere.

One future application could be the enhanced production of certain strains of vegetable or microbial biomass as a fuel source, by using a combination of exhaust gas and photosynthesis.

Of the above methods to recover the turbine exhaust heat, steam would appear to be the most attractive. This is because it is

capable of handling the large amounts of heat involved and is in a convenient form for both transportation and use as energy.

STEAM GENERATION

In the most simple form, steam can be generated using a basic fire-tube, natural circulation boiler. Increased size and complexity results from the use of an economizer and superheater in a forced circulation, water-tube boiler.

More sophisticated units generate steam at dual pressures. These permit the introduction of low pressure steam in the latter stages of a steam turbine. This reheating provides higher efficiency but at a higher cost and degree of complexity. Furthermore, in order to effect maximum efficiency, the low pressure economizer can result in a relatively low stack temperature.

To help avoid any dew point corrosion, it was decided to focus on the use of a single-stage forced circulation boiler.

Because of the large amount of excess air used for combustion and internal cooling of the turbine, the oxygen content of the exhaust is relatively high at 15-17%. This makes it suitable for firing as pre-heated combustion air in a boiler.

Using all of the available oxygen in the exhaust stream allows the production of 100,000 kg/hr of steam at 10 MPa and 510C. This would require the supplementary firing of an extra 38 MW of energy to the system with attendant increases in equipment costs. The corresponding increase in a supporting biomass infrastructure must also be considered. For these reasons, the study will investigate an un-fired boiler system.

The main piece of equipment will comprise a conventional waste heat boiler. Economizer and superheater sections will be included to achieve maximum heat recovery. Scope of supply will encompass all necessary boiler trim and auxiliaries such as re-circulation pumps, feedwater pumps, condensate pumps, deaerators, and feedwater treatment system.

For start-up and maintenance purposes, we must also include a by-pass stack and a gas-tight isolation valve upstream of the waste heat boiler. Control of the gas flow can be achieved using diverter valves in both the by-pass and boiler ducts.

Depending on the actual site conditions, an exhaust silencer may be required in the by-pass system. This is not required in the ductwork to the waste heat boiler as the boiler will provide sound attenuation and the silencer would increase the turbine back pressure. Pressure loss through the boiler and ductwork is assumed to be approximately 3 kPa under normal operation conditions.

Should the application require the production of additional electrical power, this steam can be used to drive a steam turbine generator set.

## 6.2 Air Inlet System

The air inlet system must provide for the delivery of clean air to the turbine with a minimal pressure loss. Two important functions performed by the system are filtration and silencing.

### AIR FILTRATION

The modern gas turbine requires huge amounts of air for its operation. In our study, the largest unit can require up to 7,380 Nm3 of clean, filtered air. Even 1 ppm of foreign matter in the inlet air would result in the ingestion of over 14 kg of material per day. For this reason, the control of contaminants and particulates entering with the inlet air is very important.

Filtration is necessary to prevent erosion, fouling and icing. Generally, erosion is caused by particles larger than 10 microns while fouling is caused by the smaller particles.

Ingestion of the larger particles can cause loss of performance by altering the aerodynamic characteristics of the turbine blades and stators. The trailing edges can erode to the point where they fray and break. Tip wear can open clearances and reduce the efficiency of the compressor stages.

Fouling can also impair performance by altering the turbine's aerodynamics. For any given power level, more fuel must be used with a corresponding increase in the turbine inlet temperature which can reduce the turbine life.

The inlet system must also prevent the ingestion of ice into the turbine. This ice can take the form of two different types; precipitate or condensate.

Precipitate icing results from free water drawn into the inlet. This can be in the form of rain, snow, sleet, or fog, etc. The moisture can freeze on inlet surfaces to form a build-up of ice and it is the ingestion of this ice that can cause severe damage.

Condensate icing can result from atmospheric humidity. Reduction in the inlet static pressure due to pressure losses can cause solidification of this moisture. Subsequent build-up of this ice, usually of the softer "rime" variety can cause loss of performance.

Prevention of icing usually requires heating of the inlet air. This can be accomplished directly by recirculating a portion of the turbine exhaust into the inlet air stream or the exhaust gas can be used in a gas-to-air heat exchanger.

Recirculation could be undesirable due to additional fouling of the turbine from contaminants in the exhaust gas. The use of a self-cleaning, barrier type air-filter is a possibility since it has shown to be capable of preventing both types of icing.

Any system should be equipped with blow-in doors and sensors to warn of impending actuation. This will prevent the collapse of the inlet system in the event of filter blockage.

SILENCING

Requirements for an air inlet silencer vary depending on the turbine sound characteristics and the installation site. Normally, some degree of altenuation is required.

When used, the air inlet silencer should be located as close as possible to the compressor inlet. This helps prevent resonation through the inlet ductwork.

6.3 Switchgear

The AC power switchgear provides the means for distribution of the electrical power generated by the turbine set.

The following equipment is included for the required generator protection and for control and metering purposes.

- Generator Circuit Breakers

Connects the generator to the distribution grid. The breakers open when a fault occurs in the distribution network to protect the generator. Conversely, should a fault occur within the generator itself, the breakers will open to protect the drive equipment.

The breakers are controlled by an external circuit which prevents them from closing unless the generator output is properly synchronized, i.e. both speed and phase relation are correct with the live bus or grid.

Voltage, size, and type depend upon the specific generator set and application.

- Reverse Power Relay

This device senses any condition where power in the distribution network tends to reverse direction. The relay causes the main circuit breakers to open, thus protecting the generator and associated equipment.

- Current and Potential Transformers

Provide the signals for operation of the generator control system and the electrical metering panel. This includes speed and voltage regulation circuits and operation of the synchronizing circuit along with the required frequency and phase detectors.

- Cross-current Compensation Transformer

Required when two or more generators are operating in parallel to ensure proper load sharing. Droop method is normally used to ensure proper division of the reactive load between multiple sets.

- Power Transformers

As required to provide power to the voltage regulator and series boost system, generator circuit breaker, or auxiliary package equipment.

- Current Boost Transformers

Required for the voltage regulator and series boost systems.

- Differential Transformers

Used to detect generator stator faults as a ground in one or more of the phase windings. These transformers detect a fault by comparing the current flow in the natural side of each stator phase coil to the load side of the main circuit breakers.

# P E R F O R M A N C E

## 7.1 Cycle Efficiencies

## 7.1 CYCLE EFFICIENCIES

As with any major project, the ultimate feasibility could depend on refining and optimizing various functions in order to obtain maximum efficiency.

In order to gain a better knowledge of the overall system efficiency and areas of optimization, we shall look at a number of these items individually.

OVERALL EFFICIENCY

Table 5 gives the overall performance data for the three systems evaluated with full waste heat recovery for the production of steam and electricity.

TABLE 5

Performance Data

| | SYSTEM 1 | SYSTEM 2 | SYSTEM 3 |
|---|---|---|---|
| Wood Feed kg/hr (dry) | 19,000 | 30,600 | 36,900 |
| Wood Energy MW (thermal) | 99.3 | 168.0 | 202.4 |
| Electricity production MW (net) | 28.6 | 42.0 | 52.2 |
| Overall Efficiency | 29.0 | 25.0 | 26.0 |

These overall efficiencies are for the production of electricity from green wood (50% moisture) and compare favourably with 20.1% for a conventional wood fired simple steam cycle thermal power plant and 22.6% for a complex steam cycle. (Reference 1) The complex (multi-pressure) cycle is usually only justified above 100MW electrical ouptput.

The smaller system (System 1) shows a higher efficiency in spite of the fact that under normal circumstances with conventional fuels the machine included in System 1 (Westinghouse 191) is less efficient than the BBC Type 9 or the Westinghouse 251. The reason for the higher efficiency lies in the fact that the Westinghouse 191 is capable of increasing the turbine mass flow and consequently only 5% of the compressor air must be bled from the system, all of which can be used in the gasifier.

With the larger machines (the Westinghouse 251 and BBC Type 9), 15% of the air must be bled from the system, only half of which can be utilized in the gasifier. Since it takes 2/3 of the turbine power merely to compress the air, this bleed air represents a large drain on the overall system efficiency. Any installation incorporating one of these machines should look seriously at utilizing this additional bleed air. Some possibilities are given later in this section. Utilization of the bleed air to produce additional electricity would improve the overall efficiency by one to two percentage points.

## DETERMINATION OF OPTIMUM WOOD MOISTURE

Wood, peat and most biomass feedstocks are harvested wet (or green) and contain some 50% or more water. While materials with 50% moisture can be gasified, the heating value of the gas produced is very low. In addition, more wood must be used to produce the same gasifier heat output.

In order to dry the wood, exhaust gases from the waste heat boiler can be used, and if these are not hot enough (do not contain enough total energy), turbine exhaust gas can be used to increase the temperature to the desired value. Assuming wood is delivered at 50% moisture, to dry to 30% moisture, approximately 20% of the turbine exhaust must be blended with WHB flue gas to produce sufficient drying capacity. This reduces the production of steam and electricity. To dry to 10% moisture, some 50% of the turbine exhaust must be used. Table 6 summarizes the wood consumption and electricity production at three wood moisture contents for System 3.

The following table shows that the overall efficiency does not improve by drying beyond 30% moisture. In addition, it becomes increasingly difficult to dry wood below 30% moisture requiring disproportionately long residence times. For this reason, 30% moisture has been chosen as the design wood moisture level with the drier sized to dry from 50% (as received) to 30%.

TABLE 6

| Effect of Moisture on Wood Consumption and Electricity Production | | | |
|---|---|---|---|
| Wood moisture % (w.b.) | 10 | 30 | 50 |
| Wood Consumption MW ** | 183 | 202 | 246 |
| Gas Turbine Electricity MW | 41.9 | 41.9 | 41.9 |
| Steam Electricity MW | 7.0 | 12.3 | 14.7 |
| Gasifier Air Booster MW | -0.7 | -1.0 | -1.7 |
| Misc. Electricity MW | -1.0 | -1.0 | -1.0 |
| Total Production MW | 47.2 | 52.2 | 53.9 |
| Efficiency | 25.8 | 25.8 | 21.9 |

** This is the thermal equivalent of the total wood heat input.

BLEED AIR UTILIZATION

It is understood that the system as envisioned within this report deals with a new and different turbine application. Although a number of turbines have operated on blast furnace gas, the use of a fluid-bed gasifier and wood waste or peat as a feedstock represents a non-conventional, alternative fuel source.

Gas turbines are normally designed for standard liquid or conventional fuels. They are a "constant volume" machine based on the action of the compressor blades. As such, the various components such as compressor and turbine blades and stators are designed for specific flow rates. This is known as "matching".

Because there is not a large market at present for turbines to burn this low heating value gas, we are forced to look at turbines designed for conventional fuels with their specific component matching.

In order to get the required energy input to the unit, it is necessary to inject more low heating value fuel mass than would be the case for oil or natural gas. This is in a ratio of

roughly 7:1 for lower heating value of 5 MJ/Nm3 and this additional fuel mass can represent over 15% of the turbine inlet flow.

This added mass can seriously impede turbine performance by choking the flow of gases through the turbine stages. In order to prevent this occurrence it is necessary to extract some of the air from the turbine after the compressor.

The amount of this air varies and does not necessarily equal the extra fuel mass added. This is because different turbines have different levels of loading the turbine blades and nozzles. Lightly loaded stages can tolerate larger increases without showing negative effects. For this reason, some turbines can operate on gas as low as about 7 MJ/Nm3 without having to bleed any air.

When air must be bled, it is necessary to look at possible means of using this air so that the energy it contains is not wasted.

A basic premise in the design of this system is the fact that the gasifier will operate under pressure. This pressure will be equivalent to the fuel gas pressure required at the inlet to the combustor nozzles plus the amount to overcome duct losses and pressure drops through the gas cleaning equipment.

The air for the gasifier can best be obtained from the turbine air compressor since it is already pressurized to a point near the final pressure.

Also, in order to maintain a reasonable mass balance through the turbine, some of the air must be extracted from the cycle to counteract the mass introduced by the low heating value gas. This is not as critical in the case where the turbine manufacturers have indicated modifications are possible to the turbine blades and nozzles to absorb the additional mass.

The work performed by the compressor on the air used in the gasifier can be determined as follows:

$$Wc = m\ Cp\ T1 \left( \left( \frac{P2}{P1} \right)^{\left( \frac{k-1}{k} \right)} - 1 \right)$$

Where M = mass flow, kg/sec

Cp = specific heat, of air, 1005 joules/(kg.K)

T1 = inlet temperature, (K)

$\frac{P2}{P1}$ = pressure ratio

K = ratio of specific heat 1.40

Wc = work of compression, watts

In actual fact, the efficiency of the compressor would be about 75%. This gives:

$$Wc = \frac{M}{.75} Cp\ T1 \left( \left( \frac{P2}{P1} \right)^{\left( \frac{k-1}{k} \right)} - 1 \right)$$

Using this formula and the gasifier air flow required for the various turbines, we can determine the energy savings using air compressed by the gas turbines as shown in Table 7.

TABLE 7

GASIFIER AIR POWER REQUIREMENTS

| | NATCO KG-2 | WEST 191 | BBC Type 9 | WEST 251 |
|---|---|---|---|---|
| Gasifier Air Flow Flow (kg/sec) | 1.0 | 8.65 | 13.91 | 15.62 |
| Inlet Temperature (Degrees C) | 20 | 20 | 20 | 20 |
| Pressure Ratio (P2/P1) | 3.8 | 7.48 | 8.39 | 14.0 |
| Power Required (kW) | 180 | 2,600 | 4,550 | 6,730 |

Should there still be excess bleed air over and above that required by the gasifier, this extra air can be used in other ways to recover the energy. Some of these ways are as follows:

a) Booster Power

The extra air can be used in a turbo-expander which is connected to a generator or the gasifier air booster. As this air is coming from the turbine compressor, it is heated and under pressure. By heating this air further before the expander, it is

possible to extract even more power from this air. This can be done by exchanging heat in the hot gas from the gasifier. See Table 8.

TABLE 8

BLEED AIR POWER AVAILABLE

| TURBINE MODEL | Bleed Air at Compressor Discharge Temperature | | | | Bleed Air Pre-heated to 540 degrees C | | | |
|---|---|---|---|---|---|---|---|---|
| | T1 degrees C | P1 P2 | M kg/sec | Power kW | T1 degrees C | P1 P2 | M kg/sec | Power kW |
| Natco KG-2 | 190 | 3.55 | Nil | | 540 | 3.55 | Nil | |
| West 191 | 285 | 7.01 | Nil | -- | 540 | 7.01 | Nil | -- |
| BBC Type 9 | 338 | 7.80 | 9.37 | 1917 | 540 | 7.80 | 9.37 | 2550 |
| West 251 | 390 | 13.16 | 10.34 | 2484 | 540 | 13.16 | 10.34 | 3304 |

Discharging the air to atmosphere and assuming an expander efficiency of .75, we can calculate the power produced as follows:

$$We = .75\ M\ T1\ Cp \left(1 - \left(\frac{P2}{P1}\right)^{\left(\frac{k-1}{k}\right)}\right)$$

Should we wish to avoid the extra cost of a turbo-expander, the excess bleed air could be injected into the waste heat boiler. This allows recovery of the thermal energy in the air.

For maximum efficiency, this air would be introduced at a point where the temperatures of the air and the combustion products in the boiler are approximately the same. Assuming a stack temperature of 176C, the amount of heat available in this application is as follows:

Where Available Heat = $1.05 \times 10^{-3} \times M \times (Tin - Tout)$ MW

M = kg/hr
T = degrees C

TABLE 9

| BLEED AIR HEAT AVAILABLE | | | |
|---|---|---|---|
| | Air Flow kg/hr | Inlet Temperature degrees C | Heat Available MW |
| Westinghouse 191 | 8.67 | 285 | .099 |
| BBC Type 9 | 13.94 | 340 | 240 |
| Westinghouse 251 | 15.65 | 390 | 3.52 |

This air could also be used in direct drying such as drying of the wood feed. Using a rotary dryer, for instance, the stack temperature could reduced to as low as 120C. In this case, the following heat is available.

Westinghouse 191 - 1.48 MW
BBC Type 9 - 2.95 MW
Westinghouse 251 - 4.40 MW

One further alternative would be to use this air for pneumatically conveying the wood feed. This may also effect some drying of the wood due to the close contact between the material being conveyed and the hot air.

Finally the air could be used to run an additional gasifier and produce more low heating value gas product which could then be used for exhaust gas firing or for any other process application.

INTERCOOLING

As was mentioned previously, the air required for the gasifier is best taken from the turbine compressor only if this air would have to be bled anyway. In order to achieve gas inlet pressures and overcome system losses, this air must be further boosted.

As can be seen, this power can be reduced by cooling the air prior to its introduction to the booster. Cooling the air to 80C reduces the booster power requirement to the levels shown in Table 10.

TABLE 10

| BOOSTER POWER REQUIRED | | | | |
|---|---|---|---|---|
| | Natco KG-2 | West 191 | BBC Type 9 | West 251 |
| Air Flow (kg/sec) | 1.0 | 8.65 | 13.91 | 15.62 |
| Booster Inlet Pressure (kPa abs.) | 380 | 825 | 860 | 1380 |
| Booster Inlet Temperature (degrees C) | 190 | 285 | 340 | 390 |
| Booster Outlet Pressure (kPa abs.) | 650 | 1240 | 1445 | 2135 |
| Booster Power Required kw without Intercooling | 104 | 790 | 1806 | 1838 |
| Booster Power Required kw with Cooling to 80C | 79 | 498 | 1044 | 980 |

As can be seen above, the booster power can be reduced considerably by cooling the compressor air to 80C.

Using water cooled coils, conditions for both the air and water are shown below in Table 11.

TABLE 11

| BLEED AIR COOLER CONDITIONS | | | | |
|---|---|---|---|---|
| | Natco KG-2 | West 191 | BBC Type 9 | West 251 |
| Air | | | | |
| -Flow (kg/sec) | 1.0 | 8.65 | 13.91 | 15.62 |
| -Inlet Temp. (degrees C) | 200 | 285 | 300 | 375 |
| -Discharge Temp. (degrees C) | 80 | 80 | 80 | 80 |
| -Pressure Drop (kPa) | 0.025 | 0.23 | 0.24 | 0.25 |
| Water | | | | |
| -Flow (kg/sec) | 1.89 | 5.35 | 8.93 | 8.93 |
| -Inlet Temp. (degrees C) | 21 | 21 | 21 | 21 |
| -Discharge Temp. * (degrees C) | 37.8 | 29.4 | 104.4 | 165.5 |
| -Pressure Drop (kPa) | 7.16 | 24.5 | 47.8 | 50.8 |

* Water side must be pressurized or a heat transfer oil used.

In a total energy system, this hot water could then be used for space or process heating such as pre-heating of combustion air or boiler feedwater.

Some power is required for pumping of the water but this is minimal compared to the savings in booster power.

LOAD FOLLOWING

Whilst smaller turbines can pick up additional power output almost instantaneously, the larger turbines must be controlled to increases in load of 5 to 10 % of full load per minute.

In situations where the turbine is sole supplier and having to follow load, this can produce problems of reduced voltage or "brown out". In most instances, however, the turbine would be connected to an electrical power grid and would not be required to increase power output suddenly.

If the turbine increases power output, it must draw more fuel gas. Assuming a rate of 10% per minute, calculations on the gasifier system show that the pressure of the gas would be drawn down by 70 kPa in some 20 to 30 seconds which is the time it would take for the gasifier to respond to varying load. Thus the control system should be designed to allow 70 kPa variations in low heating value gas pressure.

HEATING VALUE FLUCTUATIONS IN THE FUEL GAS

Fluctuations in the heating value of the gas must be examined to determine their impact on operation of the turbine generator set. Severe transients can cause a shutdown of the set either from high turbine inlet temperatures or possible flameout. Smaller transient can cause frequency fluctuations in the generated electrical power when not connected to a large utility grid.

As a measure of the heating value of the gas, the Wobbe index, as mentioned previously, can be used. For best operation, variations in the index should not exceed 5%. Some manufacturers specify 2%, while others allow 10% at a rate not exceeding 0.5% per second.

Experience has shown that steady-state cperation of the gasifier will produce gas near this range. Testing of the gas from a low pressure gasifier using an on-line calorimeter produced results showing a variation of approximately 6% over a six hour test period. Rate of change was 0.1% per second.

These satisfactory results were due to the consistency of feed and inherent volume of gas in the system which allowed thorough mixing.

# E C O N O M I C S

## 8. ECONOMICS

Equipment has been sized and priced for the complete system for three different sizes nominally 20 MW, 30 MW, and 40 MW of gas turbine power output.

The prices include a complete installation from wood unloading to waste heat recovery and steam turbine/generator sets.

In order to obtain a realistic range of prices and sizing, three gas turbines were chosen that are suitable for operation on low heating value gas with minor modifications.

They are the Westinghouse 191 turbine with an output of 21.7 MW, the Brown Boveri Corportaion Type 9 with an output of 30.9 MW and the Westinghouse 251 with an output of 41.9 MW. These turbines have been incorporated in systems 1, 2, and 3 respectively.

### CAPITAL COSTS

Table 12 gives a capital cost breakdown for the three systems under consideration. Costs are in 1983 dollars and are for the installed systems including engineering.

TABLE 12

Capital Cost Breakdown

| | SYSTEM 1 | SYSTEM 2 | SYSTEM 3 |
|---|---|---|---|
| | (US $ Thousands) | | |
| Site preparation | 400 | 580 | 640 |
| Civil | 500 | 730 | 800 |
| Wood Handling | 4,200 | 6,200 | 6,700 |
| Gasifier | 4,510 | 6,270 | 7,200 |
| Gas Cleanup | 800 | 1,356 | 1,321 |
| Turbine/Generator | 8,600 | 13,370 | 14,950 |
| Waste Heat Recovery | 2,600 | 3,900 | 3,900 |
| Steam Turboset | 4,020 | 5,880 | 5,520 |
| Working Capital | 750 | 1,100 | 1,200 |
| TOTAL | 26,380 | 39,386 | 42,231 |
| TOTAL WITHOUT STEAM TURBINES | 22,360 | 33,506 | 36,711 |

Table 13 is the expected performance of the three systems:

TABLE 13

| | Performance | | |
|---|---|---|---|
| | SYSTEM 1 | SYSTEM 2 | SYSTEM 3 |
| Wood Consumption kg/hr (dry) | 18,950 | 67,400 | 36,825 |
| Turbine Power MW | 21.0 | 30.9 | 41.9 |
| Steam Production kg/hr | 47,600 | 72,600 | 68,000 |
| Steam Turbine Electricity MW | 8.6 | 13.1 | 12.3 |
| Electricity Consumption MW | 1.0 | 2.0 | 2.0 |
| Capital Cost $/kW Capacity | 922 | 937 | 809 |

Using this capital cost and performance data, an economic model has been operated to calculate the discount rate that must be applied to the annual cash flows over the life of a project to achieve a Net Present Value (NPV) of zero, or

discounted net income = capital outlay.

This discount rate is the internal rate of return (IRR). The model can take any values for the cost of wood, electricity, steam, project life, rate of inflation, etc., and calculate the IRR for either an electricity only option (with electricity from both gas and steam turbines) or a combined heat and power cycle (CHP) where electricity and steam are produced as dual revenue generating streams.

Table 14 is an example of a computer printout for a typical project. It includes the smaller (System 1) package with maximum electricity production. The cash flows are done in constant dollars (1983). This gives an IRR of 17.9% before tax. If the revenue and cost streams are inflated this would increase. A 10% per annum inflation rate gives a DCFRR of 29.7%.

The prices used are for a "Base Case" scenario as given in Table 15.

Whilst electricity is normally less than 6c/kWh in most of Canada; in the United States and most of the rest of the world this price or even 10c/kWh is not uncommon. Obviously these systems will not be installed where electricity is cheap and for this reason the value of 6c/kWh has been used in this base case scenario.

## TABLE 14

DISCOUNTED CASH FLOW ANALYSIS OF GASIFIER/GAS TURBINE SYSTEM
SYSTEM 1 WITH ELECTRICITY ONLY - BASE CASE SENARIO

CAPITAL COST $ 24.65 MILLION
WOOD CONSUMPTION 19.0 TONNE/HR AT $28.00 PER TONNE
ELECT. OUTPUT 28600 KW AT $0.060 PER KW-HR

| YEAR | 1985 | 1986 | 1987 | 1988 | 1989 | 1990 | 1991 | 1992 | 1993 | 1994 |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | (1983 $ THOUSANDS) | | | | | | |
| REVENUES | | | | | | | | | | |
| ELECTRICITY | 8237 | 9610 | 10982 | 10982 | 10982 | 10982 | 10982 | 10982 | 10982 | 10982 |
| ANNUAL COSTS | | | | | | | | | | |
| WOOD | 2547 | 2971 | 3396 | 3396 | 3396 | 3396 | 3396 | 3396 | 3396 | 3396 |
| OPERATING & MAIN | 1360 | 1445 | 1530 | 1530 | 1530 | 1530 | 1530 | 1530 | 1530 | 1530 |
| UTILITIES | 102 | 119 | 136 | 136 | 136 | 136 | 136 | 136 | 136 | 136 |
| TOTAL COSTS | 4009 | 4535 | 5062 | 5062 | 5062 | 5062 | 5062 | 5062 | 5062 | 5062 |
| NET INCOME | 4228 | 5074 | 5921 | 5921 | 5921 | 5921 | 5921 | 5921 | 5921 | 5921 |
| DISCOUNTED INC | 3586 | 3650 | 3612 | 3063 | 2598 | 2203 | 1869 | 1585 | 1344 | 1140 |
| INTERNAL RATE OF RETURN | | 17.9 % | | | | | | | | |

TABLE 15

| Base Case Scenario | |
|---|---|
| Electricity | 6 cents/kWh |
| Wood | $28/ton (dry) |
| Steam | $11/ton |
| Capacity | 60% first year<br>70% second year<br>80% thereafter |
| Scheduled Operation | 8,000 hours/year |
| Life | 10 years |
| Capital Cost | $24.65 million |
| Electricity production | 28.6MW (net) |

Wood can be priced anywhere from a negative disposal cost to $60/ton for specially harvested whole tree chips. However, large quantities of wood residues are produced in North American and are used extensively as an energy resource. The open market price for this wood has tended to be in the $15 to $30 per dry tonne range. $28/tonne is then a reasonable price to use in the base case scenario.

Steam for the most part is produced from natural gas and the value of $11/tonne reflects a conservative natural gas price of $3.50/GJ plus $1.50/ton processing charge to cover boiler operating and capital costs.

The capacities of 60% in the first year, 70% in the second and 80% thereafter anticipate some shakedown problems in the first two years followed by continuous base load operation. To the extent that capacities of 90 or 95% are possible in this type of utility operation, these are conservative values. The capacities are percentages of a scheduled 8,000 hr/year (333 days/year) operation with 22 days/year for scheduled maintenance to last at least 20 years.

The cost of capital in constant dollars can be considered to be of the order of 7%. This is the current rate of interest minus the rate of inflation.

On this basis, a 17.9% rate of return is very good, but of course, as an investment it must cover any risks involved in the project, such as technology risk, security of fuel supply, uncertainty of future pricing, etc.

Under normal financing, not all the capital in a project of this nature would be equity, a substantial portion would be debt which could be serviced at the 7% rate. This means that the return on equity (ROE) would see a return much greater than the return on total investment (ROI) 17.9%, the actual rate depending on the debt to equity ratio. 80% debt would give an ROE on the 20% equity in the order of 50%. 50/50 debt to equity ratio would give an ROE of approximately 30%.

Thus under these price scenarios, an investment in this type of installation would be very attractive provided, of course, that the risks are minimized. These prices are typical for many locations in the United States.

In Canada, however, for the most part, electricity would be priced at around 3 c/kWh. With wood at $28/tonne this does not make the investment attractive. Even with wood at $8/dry tonne (there are locations with waste wood available at zero cost where $8/tonne is the cost of transportation and handling) the return is 7% which is enough only to service debt and would not pay a sufficient return on investment.

The return is attractive in Canada in an industrial location where the steam produced in a waste heat recovery boiler is used by the industry at a price of $11 per thousand kg. (Equivalent to a natural gas price of $3.50/GJ plus processing costs) and is not used to generate more electricity. In this instance, with wood at $8/dry tonne and electricity at 3.0c/kWh the Internal Rate of Return is 17.6%. The discounted cash flow analysis for this is shown in Table 16.

Another scenario in which a system would be economic in Canada is a smaller system in a remote location. The small Natco KG-2 turbine would produce a net 1.3 MW with a capital investment of $3.33 millon (see Demonstration Facility). Wood at $8/dry ton

TABLE 16

DISCOUNTED CASH FLOW ANALYSIS OF GASIFIER/GAS TURBINE SYSTEM
SYSTEM 1 WITH COMBINED STEAM AND POWER

| | | | |
|---|---|---|---|
| CAPITAL COST | $ 20.63 MILLION | | |
| WOOD CONSUMPTION | 19.0 TONNE/HR | AT | $8.00 PER TONNE |
| ELECT. OUTPUT | 20600 KW | AT | $0.030 PER KW-HR |
| STEAM OUTPUT | 48 TONNE/HR | AT | $11.00 PER TONNE |

| YEAR | 1985 | 1986 | 1987 | 1988 | 1989 | 1990 | 1991 | 1992 | 1993 | 1994 |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | (1983 $ THOUSANDS) | | | | | | |
| REVENUES | | | | | | | | | | |
| ELECTRICITY | 2966 | 3461 | 3955 | 3955 | 3955 | 3955 | 3955 | 3955 | 3955 | 3955 |
| STEAM | 2513 | 2932 | 3351 | 3351 | 3351 | 3351 | 3351 | 3351 | 3351 | 3351 |
| TOTAL REVENUE | 5480 | 6393 | 7306 | 7306 | 7306 | 7306 | 7306 | 7306 | 7306 | 7306 |
| ANNUAL COSTS | | | | | | | | | | |
| WOOD | 728 | 849 | 970 | 970 | 970 | 970 | 970 | 970 | 970 | 970 |
| OPERATING & MAIN | 1176 | 1250 | 1323 | 1323 | 1323 | 1323 | 1323 | 1323 | 1323 | 1323 |
| UTILITIES | 90 | 105 | 120 | 120 | 120 | 120 | 120 | 120 | 120 | 120 |
| TOTAL COSTS | 1994 | 2203 | 2413 | 2413 | 2413 | 2413 | 2413 | 2413 | 2413 | 2413 |
| NET INCOME | 3486 | 4190 | 4893 | 4893 | 4893 | 4893 | 4893 | 4893 | 4893 | 4893 |
| DISCOUNTED INC | 2965 | 3031 | 3011 | 2562 | 2179 | 1853 | 1576 | 1341 | 1141 | 970 |
| INTERNAL RATE OF RETURN | | 17.6 % | | | | | | | | |

and electricity at 15 c/kWh results in a IRR of 21% as shown in Table 17.

OTHER SYSTEMS

Under the base case condition of 6c/kWh, $28/dry ton wood, $11/ton steam, all the three systems show very favourable returns for both all electricity and combined steam and electricity options. Table 18 outlines the IRR for each System.

TABLE 18

| INTERNAL RATE OF RETURN | | |
|---|---|---|
| | Electricity only (%) | Electricity and steam (%) |
| System 1 | 17.9 | 25.2 |
| System 2 | 15.0 | 21.4 |
| System 3 | 21.1 | 27.0 |

The best returns are realized for the large System 3 incorporating the Westinghouse 251 turbine although the smaller System 1 with the Westinghouse 191 compares quite favourably.

The larger System 3 requires 880 tonne/day dry wood or 1,760 tonne/day wet. This is equivalent to the consumption of a medium sized pulp mill and consequently it is unlikely at this stage that there are many locations at which such a system could be installed to run entirely on wood. It is possible to run a hybrid system with natural gas that would enable the favourable economies of scale of the large system. Alternately peat could be mined and would be available in these quantities at a single location.

In all these systems, the combined steam and elctricity option shows a higher return than electricity only. This is because steam at $11/tonne generates more revenue than the equivalent electricity at 6c/kWh. Electricity would have to be 7.5 cents/kWh to make it as attractive as steam.

## TABLE 17

DISCOUNTED CASH FLOW ANALYSIS OF GASIFIER/GAS TURBINE SYSTEM
NATCO KG-2 TURBINE

CAPITAL COST $ 3.33 MILLION
WOOD CONSUMPTION 2.2 TONNE/HR AT $8.00 PER TONNE
ELECT. OUTPUT 1300 KW AT $0.150 PER KW-HR

| YEAR | 1985 | 1986 | 1987 | 1988 | 1989 | 1990 | 1991 | 1992 | 1993 | 1994 |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | (1983 $ THOUSANDS) | | | | | | |
| REVENUES | | | | | | | | | | |
| ELECTRICITY | 936 | 1092 | 1248 | 1248 | 1248 | 1248 | 1248 | 1248 | 1248 | 1248 |
| ANNUAL COSTS | | | | | | | | | | |
| WOOD | 84 | 98 | 112 | 112 | 112 | 112 | 112 | 112 | 112 | 112 |
| OPERATING & MAIN | 200 | 213 | 225 | 225 | 225 | 225 | 225 | 225 | 225 | 225 |
| UTILITIES | 12 | 14 | 16 | 16 | 16 | 16 | 16 | 16 | 16 | 16 |
| TOTAL COSTS | 296 | 325 | 353 | 353 | 353 | 353 | 353 | 353 | 353 | 353 |
| NET INCOME | 640 | 767 | 895 | 895 | 895 | 895 | 895 | 895 | 895 | 895 |
| DISCOUNTED INC | 529 | 524 | 505 | 417 | 345 | 285 | 236 | 195 | 161 | 133 |

INTERNAL RATE OF RETURN 21.0 %

## SENSITIVITY ANALYSIS

The sensitivity of all the cost assumptions have been analysed for deviations to the base case scenario as shown in Table 15.

The variations with electricity and wood cost are shown in the following graph. (Figure 14) Note the very sharp rise in the return with increasing electricity price. The curve passes through the 7% return at 5 cents/kWh indicating that to be the cost of production of electricity under this scenario.

The sensitivity to capacity and capital are shown in Table 19.

TABLE 19

SENSITIVITY OF IRR TO PLANT CAPACITY FACTOR AND TO CAPITAL COST

| | -10% | 0 | +10% |
|---|---|---|---|
| Capacity Factor | 13.7 | 17.9 | 21.9 |
| Capital Cost | 20.5 | 17.9 | 15.4 |

As can be seen, increasing the utilization by 10% (from 80 to 90%) would increase the IRR by 4%. An increase in the capital cost by 10% would decrease the IRR 2.5%.

The economic life of the plant also affects the rate of return especially for a life shorter than 10 years. Whilst a life of ten years (base case) gives a return of 17.9%, five years would reduce that to 3% and 20 years would increase it slightly to 21.8%.

In conclusion, it can be noted that generally speaking, in order for an investment to be attractive, electricity must be valued at 6c/kWh or greater, or with electricity at 3c/kWh or greater, or with electricity at 3c/kWh, there should be an industrial use for the steam generated replacing steam generated from oil or natural gas at current prices.

SENSITIVITY OF IRR TO
ELECTRICITY AND WOOD PRICES.

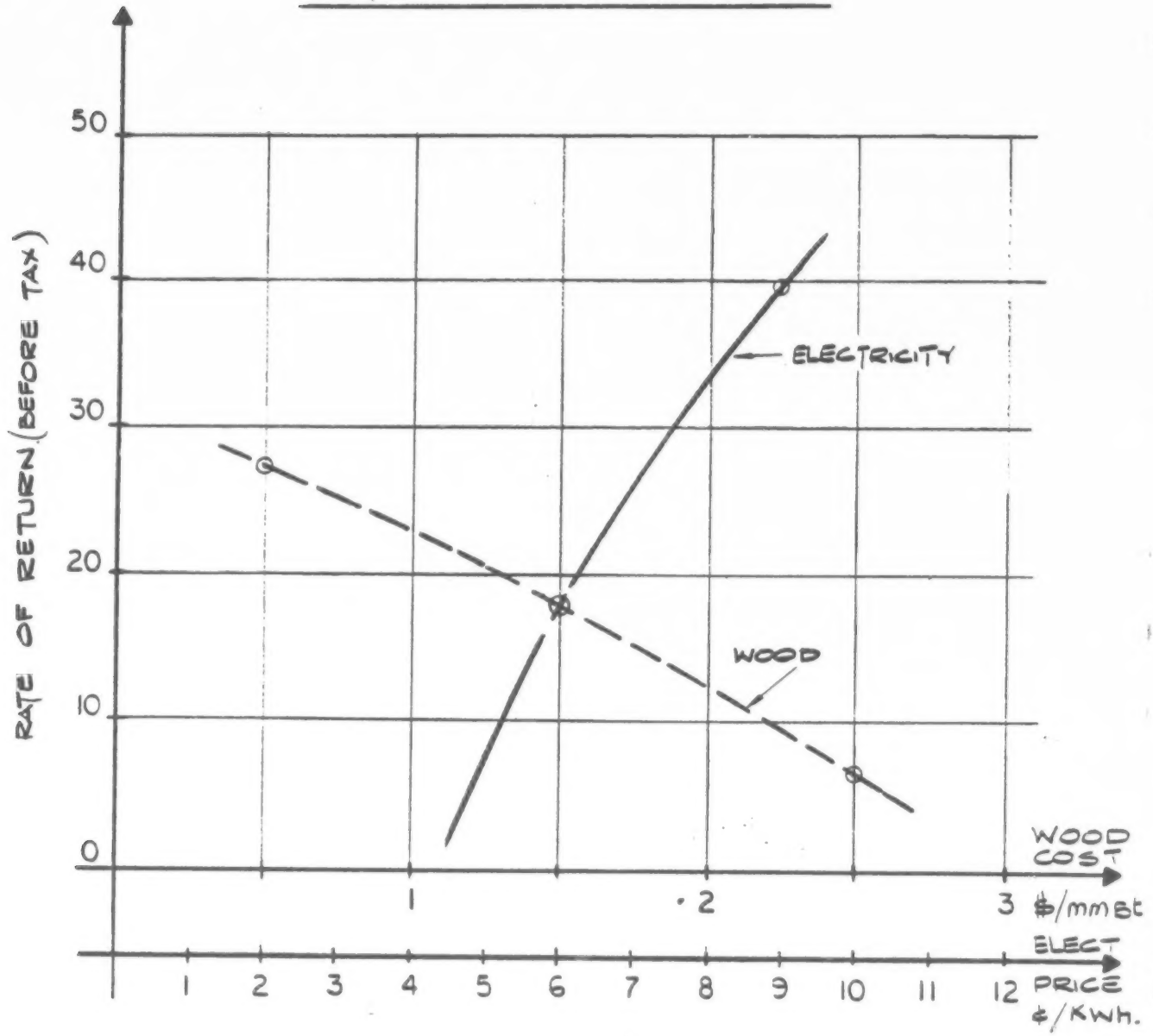



FIGURE 14

# D E M O N S T R A T I O N

# F A C I L I T Y

## 9. DEMONSTRATION FACILITY

This report has attempted to identify many of the facets of using biomass generated gas to produce electricity from a gas turbine.

This technology offers the potential for a cost effective system, divorced from normal dependance on conventional hydrocarbon fuels. Many of the components have been proven in similar, related service such as with blast furnace gas. The positive response of the turbine manufacturers, in particular, encourages further development of this technology.

On the other hand there are certain aspects peculiar to the gasification of wood or peat. Whilst equipment is available for cleaning the gas sufficiently for introduction to a gas turbine, this must still be proven on a commercial scale. The current high level of effort into hot gas cleaning technology and into turbines more resistant to contaminants, will reduce any technical risk associated with this system.

The proper selection of equipment now available will allow the construction of a demonstration facility, capable of proving and enhancing the state-of-the art in these areas.

In this regard, we believe there are two possible approaches which could be taken:

1) Large Utility-Type Facility

The first type of facility would incorporate one of the larger turbines used in our study. These units have all been well proven over many years of use on conventional fuels.

Given the right combination of fuel prices and revenue from the electricity generated, the turbine generator set could pay for itself utilizing oil or natural gas. The gasifier and related equipment, in the meantime, sized to match the turbine facility, would be built and proven to perform to specifications. The gasifier and turbine would then be matched to form a functional, cost-effective system.

Proving of the gasification system would include the gas cleaning equipment. This would ensure the production of clean gas, delivered to the specifications of the turbine.

The costs of such a facility incorporating the Westinghouse 191 turbine are shown in Table 20.

In this case, the turbine heat recovery portion of the package would be financed from the sale of heat and power generated (in the initial stages from natural gas). The major operating cost would be the purchase of natural gas. Financial burden would then shift to the production of low heating value gas at a cost-effective rate.

TABLE 20

| Large Scale Demonstration | |
|---|---|
| Turbine and associated equipment | 12,150,000 |
| Gasifier, wood handling and gas clean-up | 10,210,000 |
| Total Equipment | 22,360,000 |
| Testing and operation of gasifier for one year | 5,000,000 |
| TOTAL | $27,360,000 |

The attractiveness of such a scheme lies not only in the proving of components on a large commercial scale, but the fact that such a facility could generate enough revenue to pay for itself on conventional fuels thus minimising risk.

2) Small Dedicated-Type Facility

The second possible approach consists of a smaller, less expensive turbine/gasifier system. In this case, the turbine would likely be under 5 MW in size and would not include waste heat recovery.

The turbine would be installed and tested using conventional oil or gas. This fuel train would remain in place to provide 100% operating redundancy in fuel supply. At the same time, the

gasification and clean-up equipment will be assembled and proved. Gas quality can be tested prior to use in the turbine.

Our study shows that some of the turbines available in this size range are less sensitive to contaminant and particulate levels than the larger machines. This makes them very attractive from an operations and maintenance point of view.

The costs for a demonstration of this size are shown in Table 21.

TABLE 21

Small Scale Demonstration Costs

| | | |
|---|---|---|
| Site Preparation | $ | 58,,000 |
| Wood Handling | | 608,000 |
| Gasifier | | 1,225,000 |
| Gas clean-up | | 357,000 |
| Turbine/Generator | | 910,000 |
| Civil | | 67,000 |
| TOTAL Equipment | | 3,225,000 |
| Testing - Two years | | 1,000,000 |
| TOTAL | U.S. | $4,225,000 |

It has been shown in the Economic Section (Section 8), that such a small scale system would be economical at a remote site not currently served by a central electricity grid or at most locations where diesel fuel is currently being used to generate electricity. Thus the system could be skid mounted and tested at a convenient location and once proven and operating smoothly, shipped to the remote community for commercial operation. In this way most of the capital costs of the demonstration facility could be recovered. Feedstock would be supplied locally from a forest operation or peat deposit thus making the community more energy self sufficient.

Such a demonstration would be very attractive, therefore, both from a technical and commercial point of view.

# A P P E N D I C E S

## APPENDIX 'A'

## LIST OF TURBINE MANUFACTURERS CONTACTED

AEG-KANIS
Altendorfer StraBe 39-85
D-4300 Essen 1
Germany

ALSTHOME ATLANTIQUE
Gas Turbine Department
90001 Belfort, France

BBC BROWN, BOVERI & COMPANY
CH-5401
Baden, Switzerland

JOHN BROWN ENGINEERING LTD.
Clydebank, Dunbartonshire

COOPER ROLLS
Mount Vernon, Ohio 43050

CULLEN DETROIT DIESEL ALLISON LTD.
(Kawasaki Turbines)
Burnaby, B.C. Canada

CURTISS-WRIGHT POWER SYSTEMS
One Passaic Street
Wood-Ridge, New Jersey

DETROIT DIESEL ALLISON
Gas Turbine Section
P.O. Box 894
Indianapolis, Indiana 46206

DRESSER INDUSTRIES INC.
Dresser Clark, Div.
P.O. Box 560
Olean, New York 14760

FERN ENGINEERING
(Thomassen, U.S.)
536 McArthur Blvd.
Bourne, Mass. 02532

FIAT AVIAZIONE
Via Nizza, 312-10100
P.O. Box 1389
Torino, Italy

FIAT TTG
Via Cuneo, 20
C.P. (P.O. Box) 500
10100 Torino, Italy

THE GARRETT CORPORATION
Industrial Engines and Products
P.O. Box 5217
402 So. 36th Street
Phoenix, Arizona 85010

GENERAL ELECTRIC CO.
Gas Turbine Division
Schenectady, New York
U.S.A.

GEC GAS TURBINES LIMITED
Whetstone, Leicester
England
LE8 3LH

HISPANO SUIZA
Gas Turbines Division
B.P. 60-rue Capitaine - Guynemer
92270, Bois-Columbes
France

ISHIKAWAJIMA-HARIMA HEAVY INDUSTRIES CO., LTD.
Shin-Ohtemachi Bldg. 2-1
2-chome
Ohtemachi, Chiyoda-ku
Tokyo 100
Japan

KHD LUFTFAHRTTECHNIK GMBH
6370 Oberursel
P.O. Box 246
Germany

KONGSBERG
Gas Turbines and Power Systems Div.
P.O. Box 25, N-3601
Kongsberg, Norway

MITSUI ENGINEERING & SHIP-BUILDING CO., LTD.
Power Plant Sales Department
Machinery and Plant Engineering Headquarters
6-4 Tsukiji 5 chome, Chuo-ku
Tokyo, Japan

NATCO (North American Turbine Company)
P.O. Box 40510
Houston, Texas 77040

NUOVO PIGNONE
2 Via F. Matteucci
P.O. Box 414/50100
Florence, Italy

PRATT & WHITNEY CANADA LTD.
Longueil, Quebec Canada

ROLLS-ROYCE LIMITED
Ansty, Coventry
England
CV7 9JR

RUSTON GAS TURBINES LTD.
P.O. Box 1
Lincoln, England

SOLAR TURBINES INCORPORATED
Dept. E-101
San Diego, California 92138

STAL-LAVAL TURBINE AB
S-61220 Finspong
Sweden

SULZER BROTHERS LIMITED
P.O. Box CH- 8023
Zurich, Switzerland

TURBOMECA
Bordes 64 320
Bizanos, France

TURBOTECNICA
2 Via F. Matteucci - 50100
P.O.Box 414
Florence, Italy

WESTINGHOUSE ELECTRIC CORPORATION
Combustion Turbine Systems
P.O. Box 251
Concordville, PA 19331

## APPENDIX 'B'

## CONVERSION TABLE

1 m3 = 35.3 ft3

1 g/Nm3 = 0.413 gr/scf

1 kg/hr = 2.2 lb/hr

1 tonne = 2,220 lb

1 MJ/Nm3 = 25.4 BTU/scf

1 MW = 3.41 MM BTU/hr

'C = 5/9 x ('F - 32)

1 kJ/kg = 0.43 BTU/lb

1 kPa = 4.01 in. water

1 kPa = 0.145 psi

$10\ u = 10^{-6}\ m$

Normal conditions = 0'C, 101.3 kPa (1 atm)
Standard Conditions = 60'F, 1 atm.

Unless otherwise stated, compositions are % by weight, and wood moistures are % on a wet basis (w.b).

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